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J. F. KELLY, EDITOR

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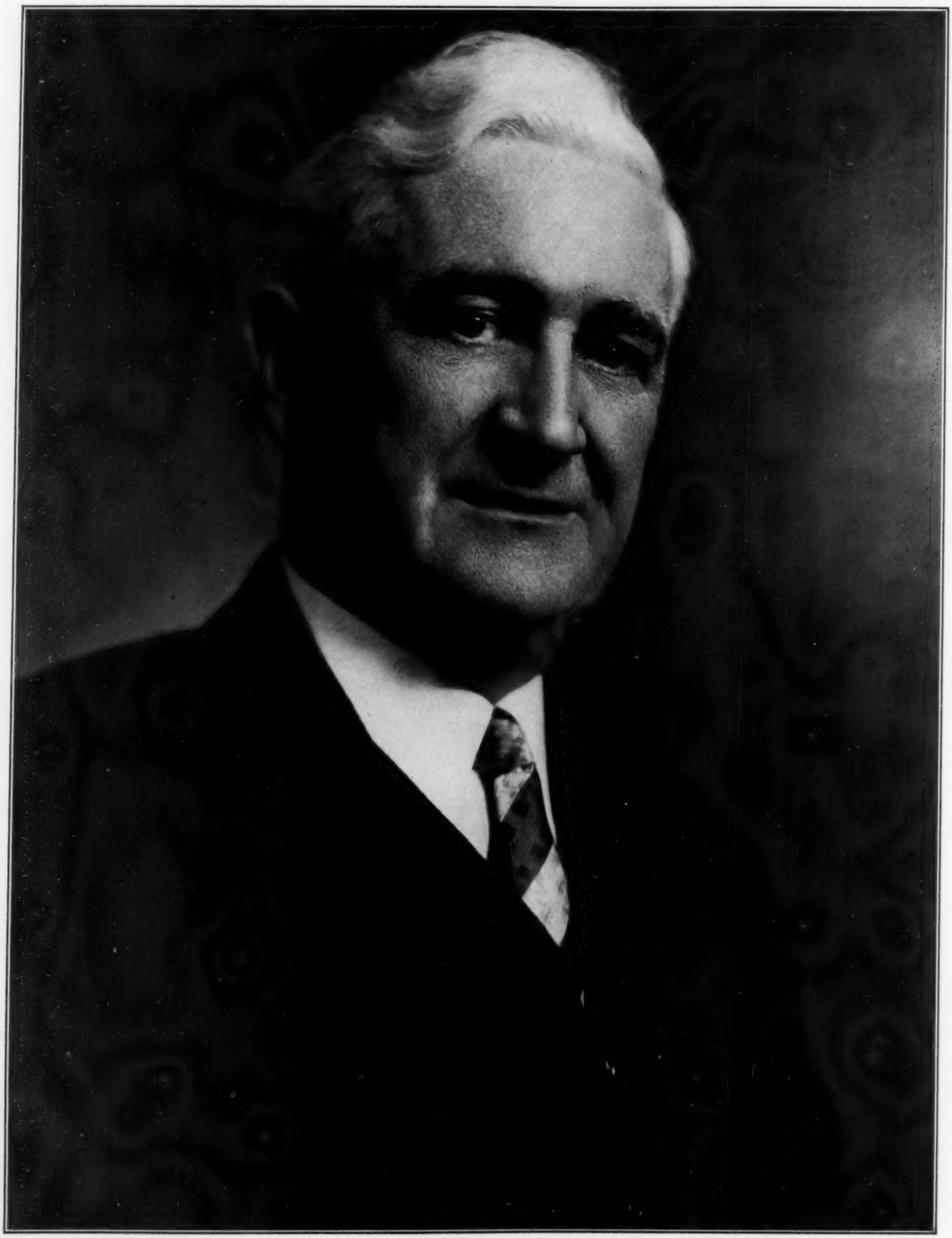

Mr. J. D. Donovan, Electrical and Mechanical Superintendent, Republic Steel Corporation, Canton, Ohio, President of the Association of Iron and Steel Electrical Engineers for the Year 1932-1933.

THE PRESIDENT'S MESSAGE

"MIGHTY OAKS FROM LITTLE ACORNS..."

Twenty-five years ago a small group of pioneers in the then comparatively new field of steel mill electrical work felt the need of some agency to foster the exchange of ideas, promote professional policies and further the development of the industry. These men planted the seed from which the present flourishing organization has grown. And even as these men met, by invitation at the plant of an electrical equipment manufacturer to pass judgment on some new apparatus, so it has remained up to the present day, for the personnel of the Association of Iron and Steel Electrical Engineers to guide and judge the many developments which have proceeded so rapidly.

Incited by the great loss of life and prevalence of accidents in the industry, a safety committee was formed in 1908 and functioned within the Association until four years later, at which time a co-operative safety meeting was sponsored, from which grew the present National Safety Council, an off-spring of which we may justly boast.

In 1921 a combustion engineering meeting was held in Chicago, and from this beginning sprung the Combustion Division, comprising at present about 250 active members, besides the associate members connected with this branch of engineering.

The year of 1925 brought safety work back as an active part of the Association program. So thoroughly does safety work permeate the activities of the steel plant engineer that it was thought that the establishment of a Safety Division within the Association would bring about the harmony and co-operation so necessary for the success of work of this kind.

The next development came in 1930 with the creation of a Lubrication Division, a logical follow-up and enlargement of work begun in various papers presented before the Association previous to this time.

The close connection of much of the work in these foregoing branches of engineering with the mechanical field led to the most recent addition, the Mechanical Division, which was formed this year and represents the outgrowth of a mechanical power transmission committee and a welding committee. This is a natural trend, as some form of mechanical power transmission is employed in almost every electrically driven unit. This close relation renders co-operation between the manufacturers of the respective equipments extremely desirable. This co-operation can be promoted by this new division, perhaps by the adoption of several sets of mechanical drive transmission specifications. There are splendid opportunities here for some excellent work, and we expect great things from this new division.

In this brief outline we see a gradual evolution towards a definite but somewhat changing goal. In all evolution, the surrounding conditions are the factors which determine the physical characteristics of the product. As these surrounding conditions change, so change the characteristics. We began with purely electrical interests and have broadened to include almost all branches of engineering in the steel industry. This evolution resulted from a definite want.

It may be pointed out that there are professional societies covering each branch of engineering. This is true, and they are excellent in their various fields. The steel industry, however, due to its size and the fact that its organization is so constituted as to demand closer relation and co-operation of the various branches of engineering, creates a need which can only be met by an organization of steel mill engineers. It is this need that we are fulfilling. We are progressing gradually, step by step. With the divisions now established we may feel that we have covered the requirements and that our organization is now complete and well-rounded. Actually, a short time may bring changed conditions which will demand further additions to our activities. When such changes occur it is necessary that we adapt ourselves to meet them. Continued resistance to such changes can only result in the fate suffered by many of the animal types of prehistoric ages. Their resistance to the process of evolution, as demanded by changing conditions, brought about their own extinction.

It is to our credit that we have thus far adapted ourselves to prevailing conditions and have moved forward. A survey of our past activities can easily be made by a study of the complete index of our Proceedings given in the January issue of the Iron and Steel Engineer for this year. Here we find a collection of valuable data covering a wide variety of subjects, entering into every branch of engineering, yet united by a common bond—iron and steel. Steel mill equipment of all types is discussed from the standpoint of design, construction, operation and maintenance, and it is this universal view-point of our work that is in a large measure responsible for much of our success and for the interest our work has aroused among steel operators outside of the Association.

Our problems are too complicated to be viewed from any one angle. There is probably no equipment unit in a steel plant into which the various branches of engineering do not enter more or less equally in design, construction, operation and maintenance, and problems involved must be surveyed from all angles. We must therefore forget that we are mechanical engineers, electrical engineers, combustion engineers, lubrication engineers and safety engineers—in short, we must be iron and steel engineers.

No mention is made, in the foregoing brief history of our Association, of the many trials and anxieties attending its development. That they were many and frequent goes without saying—nothing worthwhile is accomplished without effort. And the efforts of the founders and pioneers of our Association have borne glorious fruit. We are sound, active and vital—but this is not enough. We cannot rest on our oars and consider the pull finished. We have increased the scope of our activities, and in doing so, we have increased our responsibilities. We must carry on our new work with the same effort and vigor that was put into the building of the Association in the early lean years, so that the results shown will uphold the excellent reputation achieved in the past. To do this requires the co-operation of every man in the organization. A little effort on the part of every individual will produce the desired results, while the burden of each is light. Let us all give this little push, and the resulting momentum will carry us on to new heights of achievement. We must progress, that we may not fall back.

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Water-Cooled Furnaces Steam Boilers

By
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Presented before Combustion Engineering Division of the Association of Iron & Steel Electrical Engineers at the Twenty-eighth Convention in Pittsburgh, June 22, 1932.

Probably no part of the steam generating plant has been discussed as much in the past few years as the boiler furnace. Considering the fact that satisfactory furnaces have been in use for many years, an examination of the requirements past and present will lead to a better understanding of the elements involved in our present day constructions.

A furnace is an enclosed space, hence, the problem of proper furnace design resolves itself into the problem of designing walls or boundaries: first, as they relate to the position and form: second, as they affect maintenance, and third, as they affect combustion.

Before describing furnaces that have been used, it may be well to outline the requirements to be met, as with these in mind various furnaces can be compared and evaluated to better advantage.

The requirements are:

1. Minimum total cost, taking into account both the life of the furnace and the cost of maintenance .
2. Insurance of safety and continuous operation.
3. Tightness against air leakage inward and gas leakage outward.
4. Maximum rate of complete combustion with minimum excess air without exceeding the limits imposed by the deposition of slag on the walls, in the ash pit and on the boiler tubes.

Fundamentally, the combustion rate is limited only by the speed with which the elements involved will combine. This rate is so high and present rates obtained so low, by comparison, that it can be considered infinitely high, or as a rough approximation it can be stated as being above 1,250,000 B. t. u. per cubic foot under certain special conditions with a flame length of the order of four inches. This limitation need not be considered here. Neither is it advisable to speculate on the possible limitations of extremely high furnace temperatures which might be obtained with a molecular mixture of fuel and air; burners must be much improved before this limitation will be serious.

The maximum rate of complete combustion, from a practical standpoint, involves the case of coal, the problem of handling the slag caused by the fusing temperature of the ash being lower than the temperature of the furnace. When the temperature at which the ash fuses is exceeded, and it must be exceeded at commercial rates of firing, three conditions are brought about:

a. Fused or semifused ash in the ash pit. With stokers this is manifest by clinker trouble on grates and on side walls and bridge walls. With fuels containing ash of low fusing temperature burned in suspension, certain precautions must be taken as described later.

b. The action of liquid ash on walls. In stoker fired furnaces, where the fuel is burned on grates at relatively low rates of combustion, this limitation did not exist a few years ago except at the sides of the fuel bed; but the more recent demand for increased capacity has placed the same limitations upon stokers as those affecting the burning of ash-containing fuels in suspension.

c. The collection of liquid or semi-liquid ash on the convection tubes of the boiler. With a great majority of ash-containing fuels having an ash fusing temperature of 2300 degrees and below, the fuel-burning capacity of furnaces is limited by the temperature and quantity of the ash leaving the furnace.

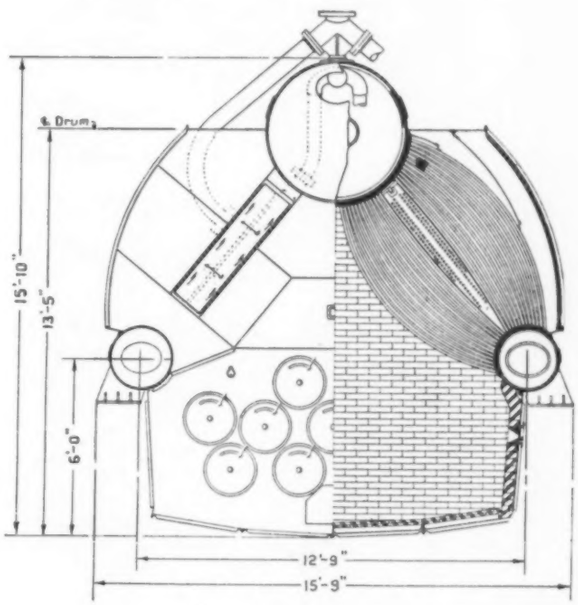

FIG. 1.

A survey of boiler furnaces installed during the last fifteen years may be of value in showing the changes in requirements and how they have been met.

In tracing these developments from 1918 we find certain stoker fired furnaces in which there was some trouble experienced due to clinker adhesions on side walls and bridge walls, but which nevertheless met the requirements fairly well. Examination of some of these furnaces and the operating conditions of that period provides a suitable starting point from which to follow the changes in requirements which have taken place.

Figure 1 shows a B & W Express Type Marine

Boiler installed and tested by the United States Navy at the League Island Test Plant, Philadelphia, Pa., in 1918. This unit had a heating surface of 8,318 square feet, a furnace volume of 751 cubic feet and was fired by eleven B & W mechanical atomizing oil burners. At a maximum rating of 660% the rate of heat liberation in the furnace was 291,000 B. t. u. per cubic foot. Figure 2 shows the efficiency obtained with this unit throughout a wide range output.

The heat liberation in this unit was probably the highest obtained in any boiler furnace of that size up to that time and, as a matter of fact, it is doubtful if these rates have ever been materially exceeded when using cold air with fuel oil or any other fuel with the exception of gas.

At the maximum rating given the combustion in the furnace was not complete as is shown by the falling off in the efficiency curve above 330% of rating. At 330% rating, however, the combustion in the furnace was so nearly complete that the unaccounted for loss on the unit was 1%. We, therefore, can consider this capacity as representing complete combustion.

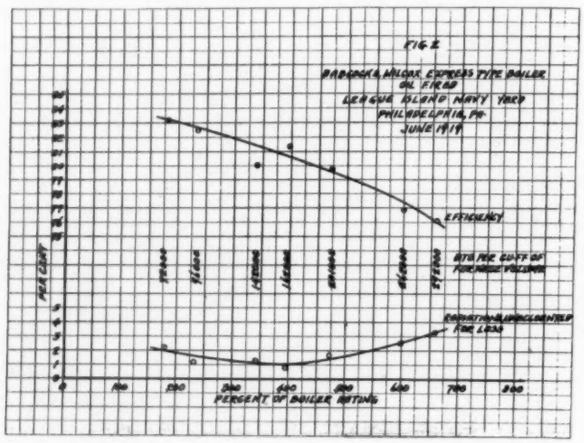

FIG. 2.

At 330% rating the heat release was 134,000 B. t. u. per cubic foot. The heat release per front foot of boiler was 4,680,000 and the B. t. u. release per square foot of cold surface exposed to the furnace was 599,000.

Since the boiler shown in Figure 1 gave such excellent results, it may be well to consider what, if anything unusual, was responsible for these results. The answer to this question is that these results were largely due to the excellent atomization of the fuel by the burner, and the introduction of air for combustion around the burners to provide an intimate mixture of the fuel and air entering the furnace. The quantity of ash in the fuel was so small that high furnace-temperatures could be maintained without encountering trouble with the refractory lining.

Figure 3 shows a side elevation of a Class W boiler at the Delray Station of The Detroit Edison Company fired by under-feed stokers and erected prior to the installation shown in Figure 1. West Virginia coal was burned having an ash fusing temperature of 2700 deg. fahr. This unit contained

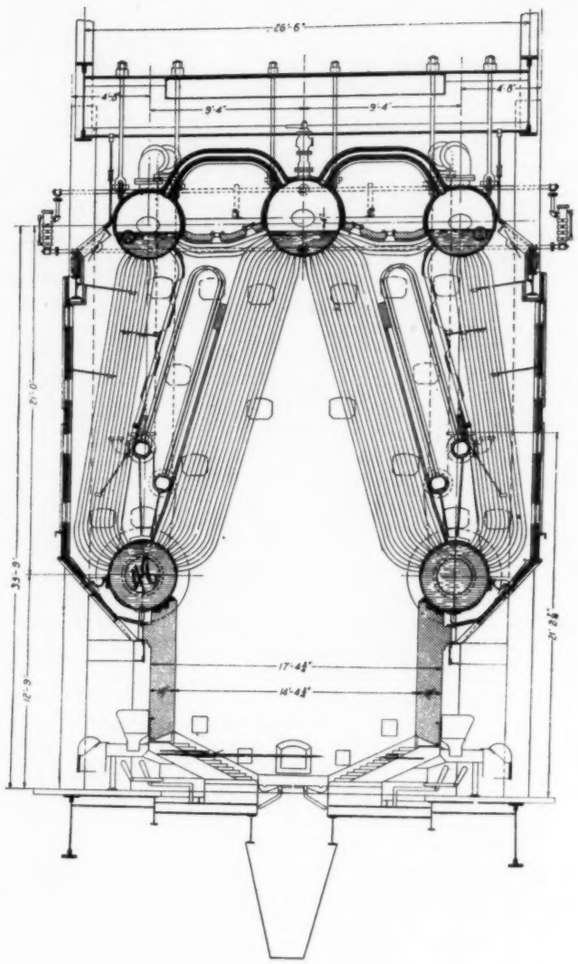

FIG. 3.

23,654 square feet of heating surface. The furnace volume was 8170 cubic feet, the unit being 26 feet wide and fired from both ends. These units operated for years with satisfactory results at a capacity of approximately 147,000 pounds of steam an hour representing up to that time probably the largest units in steam output in operation. At the capacity stated, little if any trouble with furnaces was encountered other than along the slag line so that a consideration of the furnace factors in these units lead to a measure of the principles underlying furnace design. These figures are:

B.t.u. per cubic foot	27,500
" " front foot including both sides	4,200,000
" " square foot of cold surface exposed to the furnace	393,000

Another example of furnaces in use at about this time is shown by Figure 4. These units consisted of boilers of 11,400 square feet of heating surface and operated at a capacity of 90,000 pounds of steam per hour. The furnaces were quite satisfactory and their maintenance extremely low. A consideration, therefore, of the conditions under which these furnaces were operated is of interest.

At 90,000 pounds of steam per hour the B.t.u. heat release was:

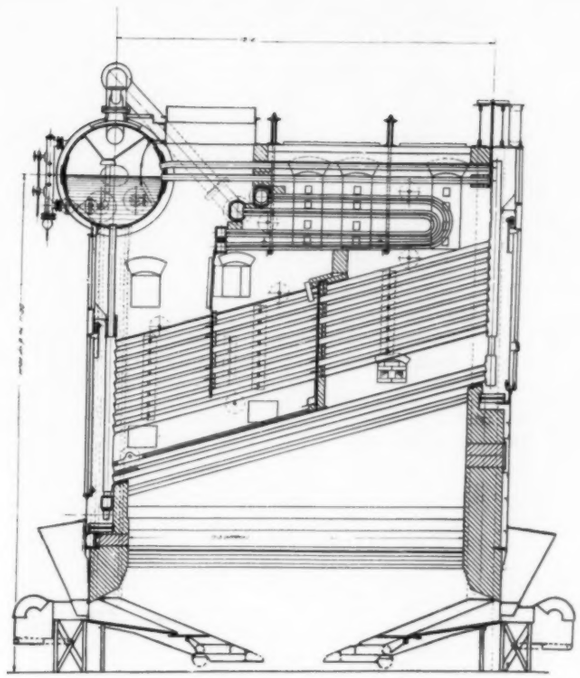

FIG. 4.

B.t.u. per cubic foot	35,000
" " front foot including both sides	2,600,000
" " square foot of cold surface exposed to the furnace	301,000

The three units described, as stated, are typical furnace designs of 1918 or thereabouts, meeting the conditions of the time with such satisfactory results that it would be difficult to imagine the necessity for radically new designs in 1932, except on the basis that the conditions of 1932 differ radically from those of 1918. A consideration of these differences should, therefore, be helpful in solving present day problems.

In 1918, after considerable work had been done by some pioneering engineers on about ten B & W boilers at the plant of The Puget Sound Traction Light & Power Company, Seattle, Washington, it was demonstrated that low grade coal could be burned in pulverized form under boilers.

Figure 5 shows the side elevation of one of these units. The pulverized coal equipment and the design of furnace were furnished by the Fuller Lehigh Company. These units were small, only 300 horsepower, and operated at capacities of about 150% of rating. The furnaces of these units were of importance mainly for the reason that it was demonstrated that pulverized coal could be successfully burned under boilers, and this fact was brought to the attention of power users and emphasized the possibilities of further development along this line. We can consider the work at Seattle as the beginning of one of the changes which has taken place between 1918 and 1932, which has had so much to do with the question of furnace design. It is needless to point out how this new tool was further advanced by the energy and foresight of the late John Anderson of Milwaukee and many other engineers of that period.

Starting with the installations that have been

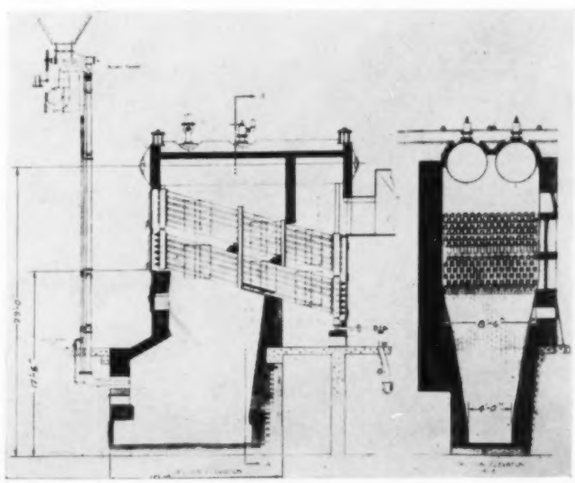

FIG. 5.

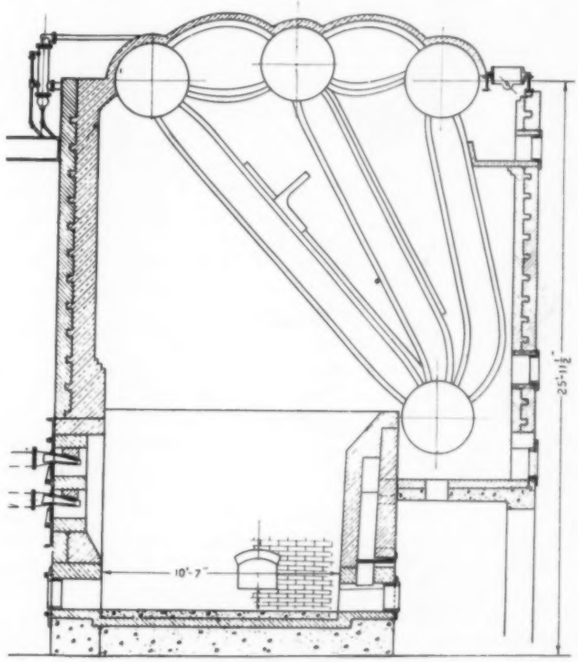

FIG. 6.

described, the demand for increase in capacity per unit began to be more or less general in the power field in about 1918. This was not new at that time as probably the vision of large size units under very careful control was seen by Alexander Dow, President of The Detroit Edison Company, as exemplified by his installation of 2200 horsepower boilers in about 1910. Apparently it took the intervening years to satisfy the general power fraternity that units of such size were practicable, and having become convinced that such was the case, engineers in the power industry demanded not only large size units but also higher relative capacities from such units, and to this desideration pulverized fuel was an excellent tool.

The effect of these two factors is perhaps most

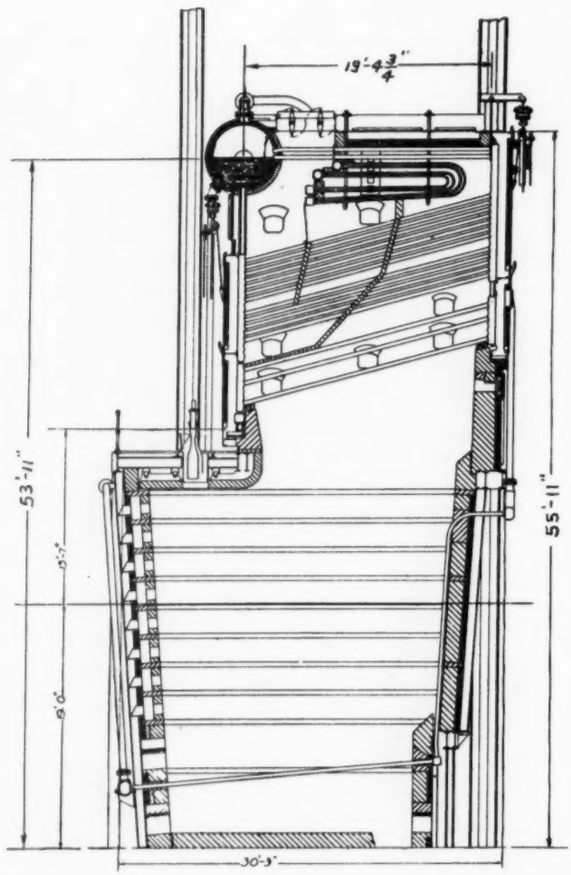

FIG. 7.

easily observed in noting the development in boiler furnaces from about that time to the present. Almost from the beginning the use of pulverized coal, particularly with normal coals having an ash fusing temperature of less than 2300 degrees, resulted in difficulties due to slag. These troubles consisted of erosion of brickwork, difficulty in removing ashes and slagging of the boiler tubes. The first efforts were directed toward making the furnaces large enough and applying the air at a low enough velocity to obtain what was commonly termed mellow flame conditions. An example of this type of unit is shown in Figure 6, showing a side elevation of the Stirling boilers and the Fuller Lehigh furnaces at The Trumbull Steel Company plant at Warren, Ohio, which, as you will note, shows horizontal burners with secondary air supplied through the bridge wall. These units operated quite satisfactorily with a heat release of 17,900 B.t.u. per cubic foot, burning a coal having an ash fusing temperature of 2400 degrees with about 30 per cent excess air.

It soon became evident that employing a relatively large furnace volume and mellow flame conditions limited the value of pulverized fuel. The development turned toward the addition of water cooling surface to the furnace floor and walls to protect brickwork and maintain furnace temperatures low enough to prevent slag difficulties. Later the development included turbulent burners which complete the combustion of the fuel in a reasonable space and improve water-cooling to permit of higher furnace temperatures than the brickwork used previously could withstand.

Figure 7 is a side elevation of No. 6 boiler at the Springdale Plant of The West Penn Power Company, installed in 1922, showing water cooling across the bottom of the furnace and on the rear wall. The boiler contained a total heating surface of 15,500 square feet and operated at a capacity of 180,000 pounds of steam per hour with the following heat release:

B.t.u. per cubic foot	20,600
" " front foot	10,250,000
" " square foot of cold surface exposed to the furnace	327,000

The progress from this period, 1922 can be easily followed at the Cahokia Station of The Union Electric Light & Power Company at St. Louis. Figure 8 shows the first units installed similar to the one

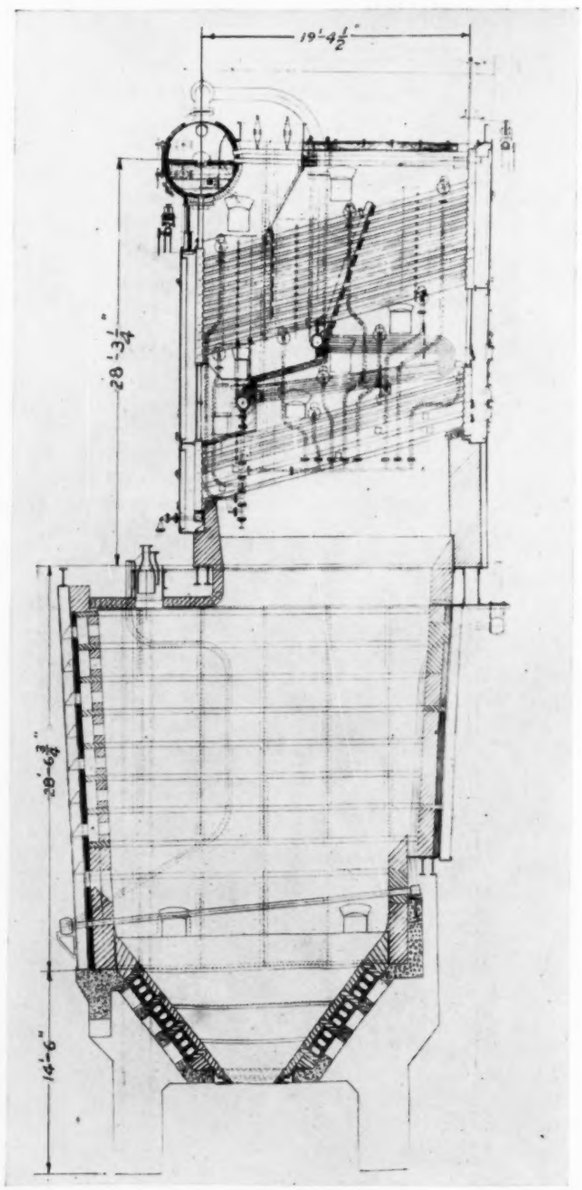

FIG. 8.

installed at Springdale and of approximately the same capacity, but using coal with an ash fusing temperature of approximately 2000 degrees. At a capacity of 129,000 pounds of steam per hour the heat release was:

B.t.u. per cubic foot	16,160
" " front foot	8,440,000
" " square foot of cold surface exposed to the furnace	222,500

Under these conditions, brickwork maintenance on the side walls proved excessive and ash removal was difficult. In the next units, installed in 1925, shown in Figure 9, the water-wall surface was increased by the addition of side-wall cooling tubes.

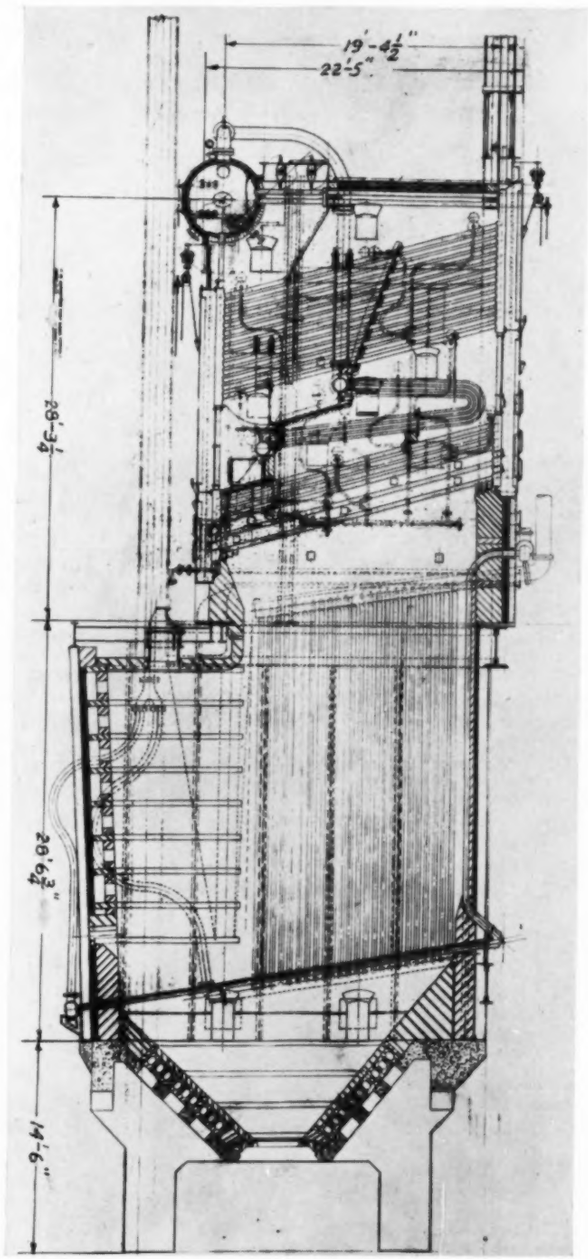

FIG. 9.

At 140,000 pounds of steam per hour the heat release was:

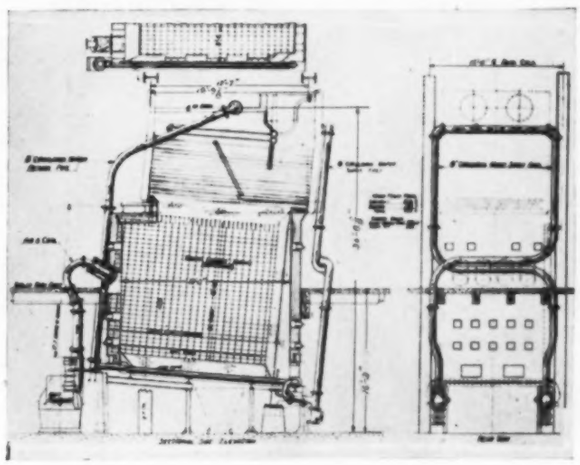

FIG. 10.

B.t.u. per cubic foot	15,400
" " front foot	9,260,000
" " square foot of cold surface exposed to the furnace	118,500

About 1925, the same company installed in their Ashley Street Plant a water-cooled furnace under a number of their older boilers. A side elevation of this installation is shown in Figure 10, Bailey refractory faced block covered tubes were used in the side walls and smooth metal blocks on the floor. Circular burners were employed in which pulverized coal was burned by admitting all of the air for combustion at the burners with a sufficient velocity to cause a quick mixing and a high speed of ignition and combustion. At the capacity of 55,000 pounds of steam per hour the heat release was:

B.t.u. per cubic foot	23,120
" " front foot	6,208,000
" " square foot of cold surface exposed to the furnace	133,300

No effort was made in this furnace to keep the slag from fusing and it, therefore, marks a turning point in the design of pulverized coal furnaces.

The last installation at the Cahokia Station, as indicated in Figure 11, shows the development at this station to a furnace with a Bailey-Block-covered hopper and surrounded by water-cooling surface except for the refractory portion around the circular burners in the front wall of the furnace. These units operated at 240,000 pounds of steam per hour, double the capacity of the first units installed four years previously. At this capacity the heat release was:

B.t.u. per cubic foot	24,200
" " front foot	13,500,000
" " square foot of cold surface exposed to the furnace	184,500

About this time the advance in capacity and the use of pulverized coal met the advance in the use of high-pressure steam in the installation of the Two Stage Boiler at the Calumet Station in Chicago. This unit was first put in operation in 1926. Figure 12 shows the side elevation of the furnace completely surrounded by Bailey-Block walls containing

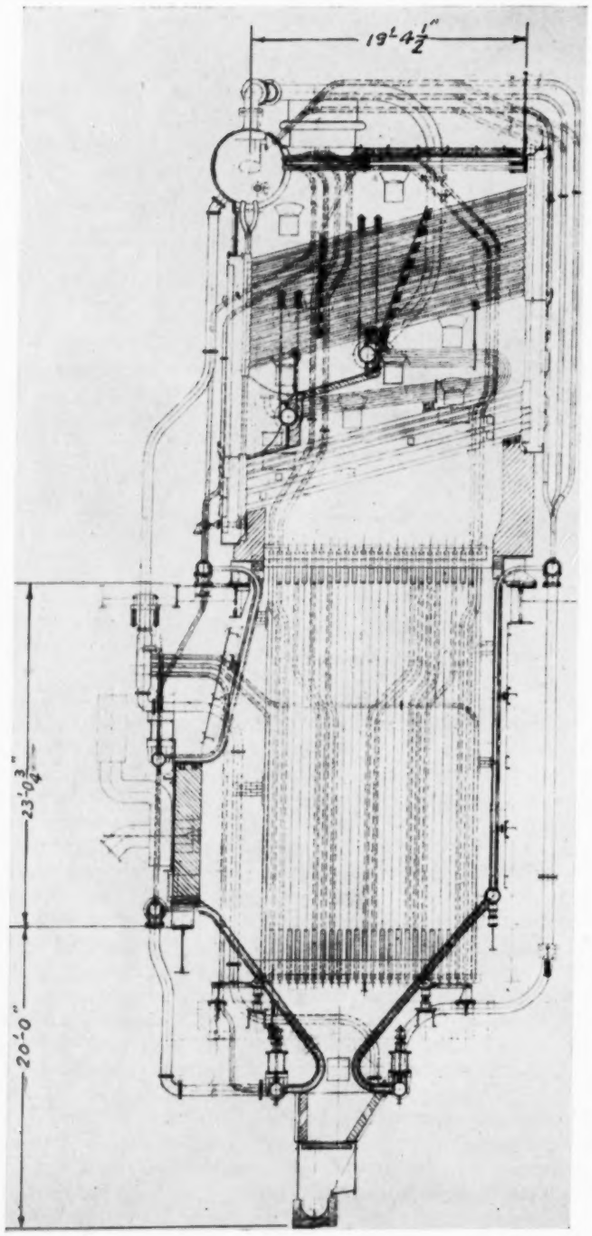

FIG. 11.

refractory faced metal blocks with the exception of the sloping floors which consisted of smooth metal blocks. The figure also shows the boiler unit with a steaming economizer and air heater in the rear. This unit was designed to produce 225,000 pounds of steam per hour and has operated at capacities as high as 290,000 pounds of steam per hour. In regular service as required for plant loads it generates 250,000 pounds of steam an hour.

Full information in regard to its efficiency in operation is reported by A. E. Grunert (1) when using both Illinois coal of 1950 deg. fahr. fusing temperature and coals of considerably higher ash fusing temperature. At 250,000 pounds of steam per hour the heat release is:

(1) Of the Commonwealth Edison Company before the A.S.M.E.

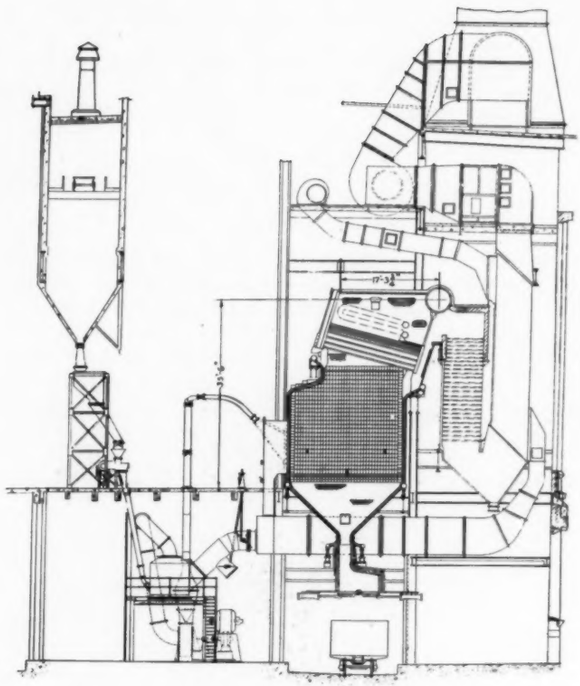

FIG. 12.

B.t.u. per cubic foot	28,200
" " front foot	14,700,000
" " square foot of cold surface exposed to the furnace	224,000

Another notable step in the development of pulverized fuel furnaces was made at The Buffalo General Electric Company in 1926 where, starting with a well type furnace eight feet square using tangential firing to develop 250,000 pounds of steam an hour, it was found necessary to change to a sixteen foot well. This development was ably covered by Cushing and Moore (2) and E. G. Bailey (3). Figure 13

(2) In their paper entitled "Direct Fired Powdered Fuel Boilers with Well Type Furnace at Charles R. Huntley Station", presented at the First National Fuels Meeting, A.S.M.E., St. Louis, Mo., October 11, 1927.

(3) In a paper entitled "Some Factors in Furnace Design for High Capacity", presented at the Annual Meeting of the A.S.M.E. in New York, December, 1927.

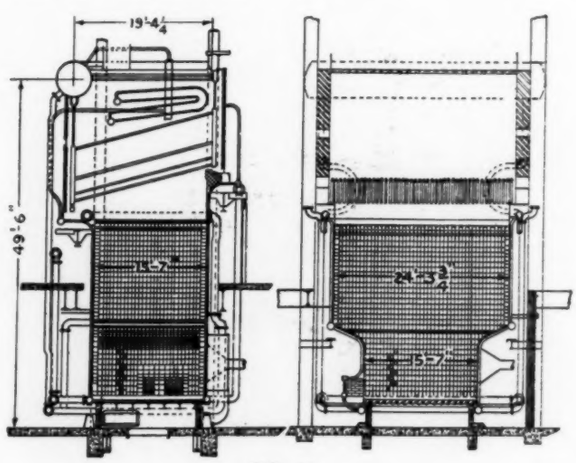

FIG. 13.

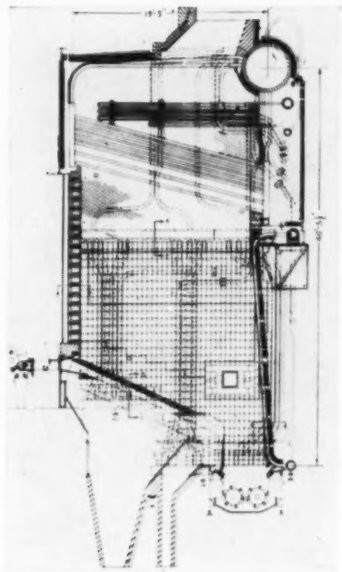

FIG. 14.

shows the front and side elevation of the sixteen foot well unit with slag tap bottom, burning coal having an ash fusing temperature of approximately

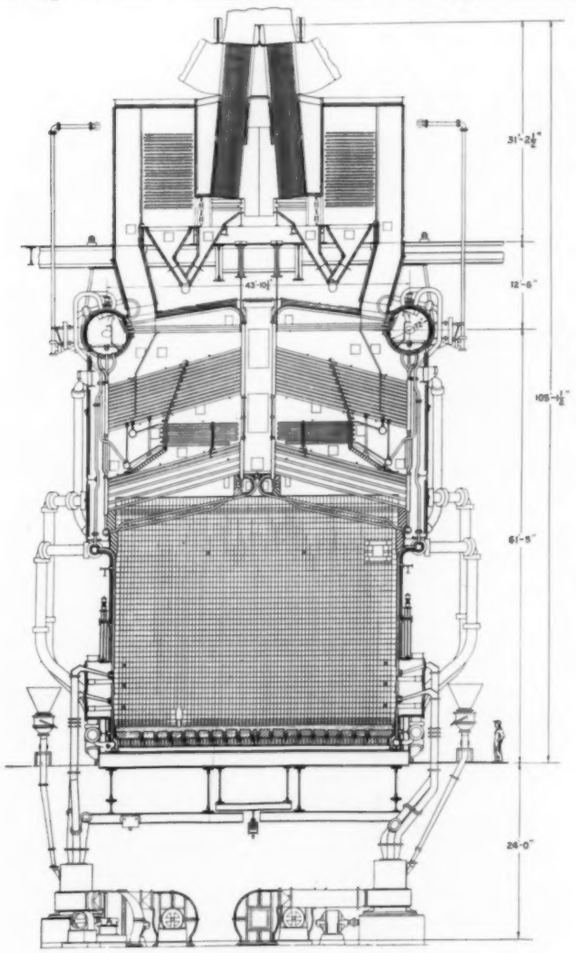

FIG. 15.

1950 degrees. These units operated at 275,000 pounds of steam per hour with a heat release of:

B.t.u. per cubic foot		40,450
" " front foot		8,486,000
" " square foot of cold surface exposed to the furnace		314,000
Excess Air	(approx.)	20%

This installation represents not only a peak for its time in burning pulverized fuel within a given volume, but represents the original disposal of the ash in liquid form.

The progress from here on consisted largely of the consolidation of the gains which had been made, and the developments in burners to improve still further the mingling and turbulent action of the fuel and air in order to burn the fuel more quickly and completely and increase the capacity and efficiency of a given unit. A brief description of some typical modern units illustrates the development of furnaces and at the same time gives some idea of the general acceptance of high pressure steam.

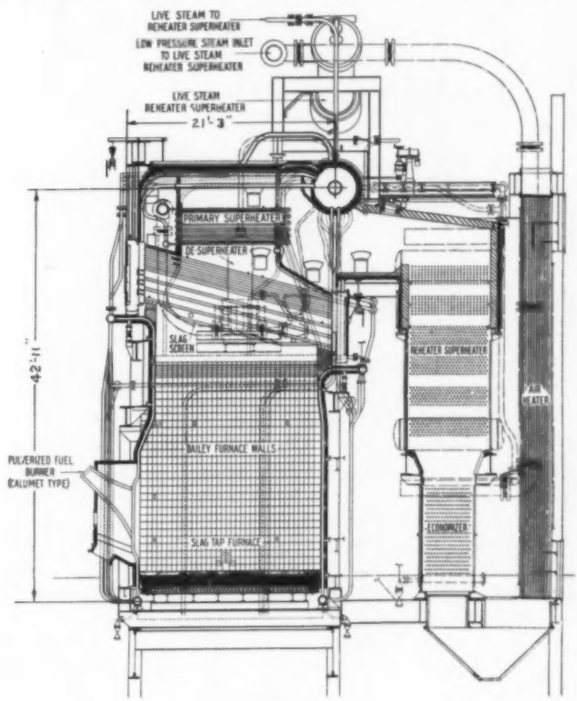

FIG. 16.

Figure 14 shows the latest two stage units installed at The Edison Electric Illuminating Company of Boston which was the first power plant to use boilers built for 1200 pounds working pressure. This development is described in a paper presented by Moultrop (4). These units are fired with underfeed stokers, which have progressed considerably in capacity since 1918 and are built for 1400 pounds pressure with block covered furnace walls on the sides and rear. At 300,000 pounds of steam the heat release is:

B.t.u. per cubic foot	42,500
" " front foot	16,000,000
" " square foot of cold surface exposed to the furnace	316,000

(4) At the Metropolitan Section of the A.S.M.E., and published by "Electrical World", March 16, 1929.

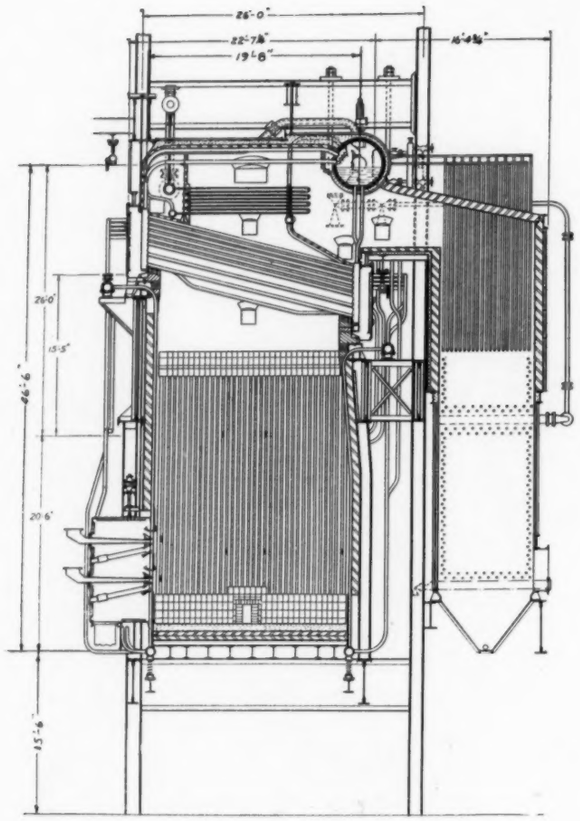

FIG. 17.

Incidentally, these units were also equipped with air heaters, which are now in common use, the air to the stokers being heated to 400 degrees further increasing the severity of furnace conditions.

Figure 15 shows the twin units installed at Hellgate Station of the United Electric Light and Power Company for 800,000 pounds of steam an hour, which have operated up to 1,000,000 pounds, this limit being entirely one of forced and induced draft fan capacity. At the capacity of 800,000 pounds of steam per hour the heat release is:

B.t.u. per cubic foot	28,200
" " front foot	16,750,000
" " square foot of cold surface exposed to the furnace	243,000

Careful tests have been made of these units showing an over all efficiency of approximately 87% at 800,000 pounds per hour and 88.7% at 400,000 pounds. Water-cooled block walls are used on all four sides of the furnace. The side walls of these units are 40 feet wide, probably the maximum of any installations made to that date.

Cross tube burners are placed in each end wall, which represents further progress along the line of more intensive burning. Slag tap floors are installed and successfully used although the coal at this plant has an ash fusing temperature of approximately 2450 degrees.

Figure 16 shows the installation of 1400 pound boilers at the South Amboy Plant of The Jersey Central Power and Light Company. The slag tap bottom with water-cooled tubes successfully withstands the action of high sulphur slag. These units operate at 280,000 pounds of steam an hour giving a heat release in the furnace of:

B.t.u. per cubic foot	27,500
" " front foot	13,440,000
" " square foot of cold surface exposed to the furnace	207,500

Figure 17 shows the side elevation of one of the units at Deepwater Station of The American Gas & Electric Company. Here again we find slag tap bottoms with cooling tubes in the walls, the bottom portion of which is covered with blocks, the remainder of the walls being protected by bare tubes. Operating at a capacity of 330,000 pounds of steam an hour the heat release in these furnaces is:

B.t.u. per cubic foot	22,600
" " front foot	16,130,000
" " square foot of cold surface exposed to the furnace	167,500

Figure 18 shows the installation of a large boiler unit at the plant of the Diamond Alkali Company at Painesville, Ohio. The boiler operates twenty-four hours a day at 250,000 pounds of steam an hour and a pressure of 750 pounds. A slag tap furnace is used with block covered cooling tubes on the walls. At the capacity stated, the heat release is:

B.t.u. per cubic foot	24,370
" " front foot	15,206,000
" " square foot of cold surface exposed to the furnace	148,000

Figure 19 shows the side elevation of one of the units installed at Delray Station No. 3 of The Detroit Edison Company. Here is shown a special Stirling boiler fired by underfeed stokers with completely water-cooled furnaces. The lower portion of the furnace walls is equipped with Bailey smooth

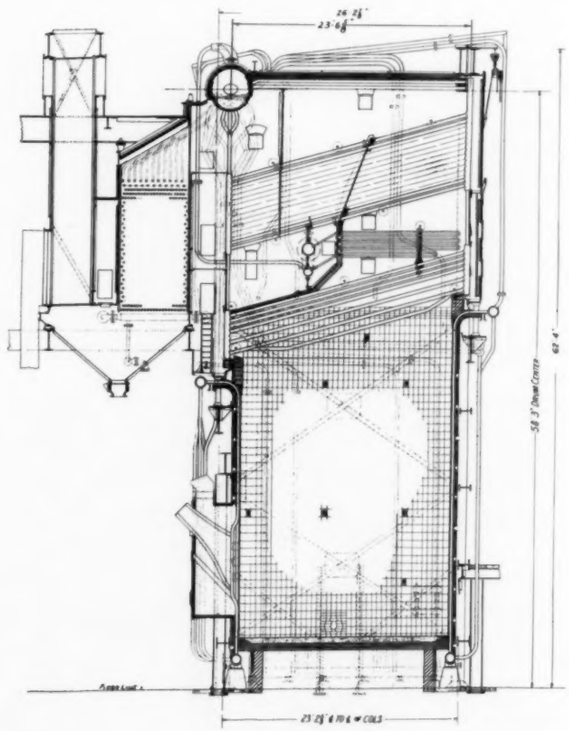

FIG. 18.

metal blocks to minimize clinker formation and withstand erosion due to intense air blast. These boilers, with furnaces twenty-six feet wide, have operated at over 400,000 pounds of steam an hour at which capacity the heat release was:

B.t.u. per cubic foot	39,800
" " front foot	20,400,000
" " square foot of cold surface exposed to the furnace	275,000

Figure 20 shows the side elevation of the last unit at Super Power Company, Pekin, Illinois. This station, as you know, holds an enviable record of low B.t.u. per kilowatt so it is of interest to note the boilers and furnaces which help make these results possible. The Babcock & Wilcox Boiler unit is equipped with block covered furnace walls, slag tap bottom and is fired by Cross-tube burners. The fuel used at this plant is Illinois coal having an ash fusing temperature of approximately 1950 degrees. These units operate at a capacity of 350,000 pounds of steam per hour though this capacity has been exceeded materially. At this capacity the heat release is:

B.t.u. per cubic foot	22,600
" " front foot	12,380,000
" " square foot of cold surface exposed to the furnace	153,500

The progress made in increasing the efficiency of large generating stations over a period of years is

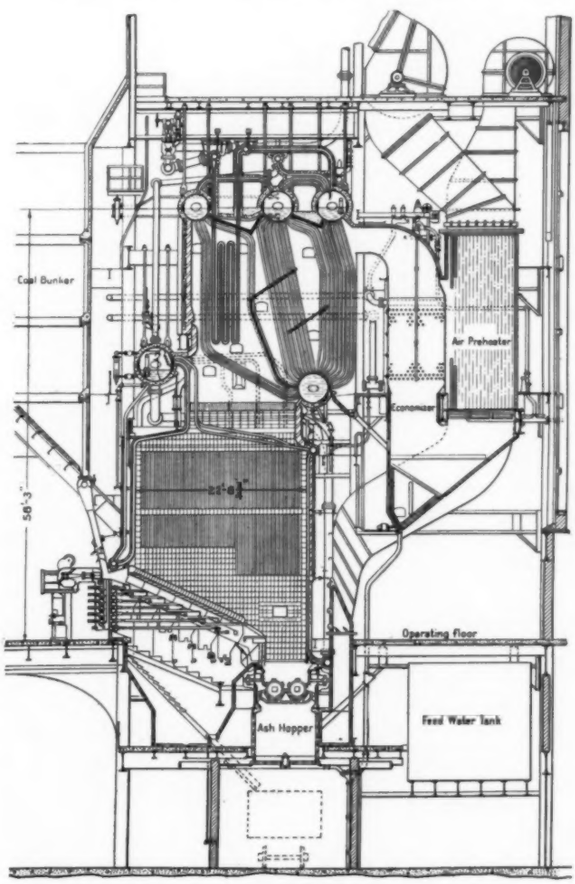

FIG. 19.

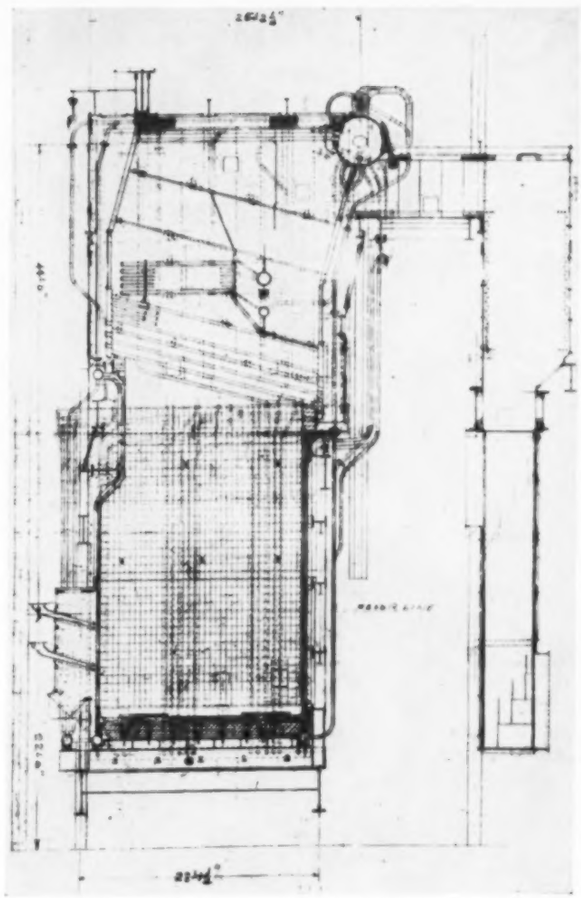

FIG. 20.

well set forth in the Presidential Address made by William S. Monroe (5).

Figure 21 shows the first units installed at the Chicago District Electric Generating Corporation, State Line Station. These units operate regularly at capacities up to 450,000 pounds of steam per hour and have operated at 500,000. Block covered walls surround the furnaces which have hopper bottoms. At 450,000 pounds of steam per hour the heat release is:

B.t.u. per cubic foot	27,500
" " front foot	16,600,000
" " square foot of cold surface exposed to the furnace	193,500

We shall refer to this station later to discuss some experiments which have been conducted and showing further improvements in furnace design.

Having shown units using some of the best coals available also units using Illinois coal having perhaps the lowest fusing temperature, used in firing boilers, we will now describe units burning natural gas or fuel oil.

Figure 22 shows the latest installation of The Southern California Edison Company, Long Beach, California. These furnaces of Bailey block construction are almost completely water-cooled. The water-

(5) To the Western Society of Engineers on May 22, 1930.

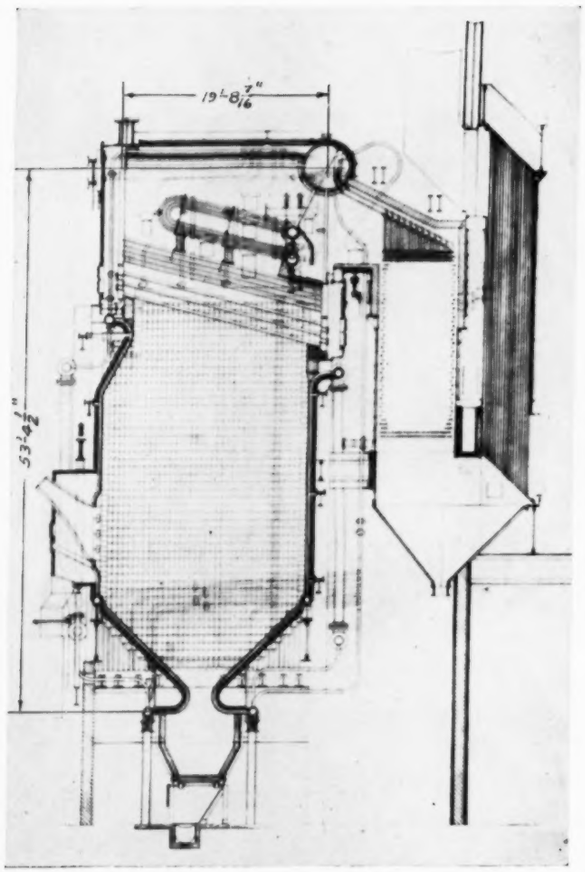

FIG. 21.

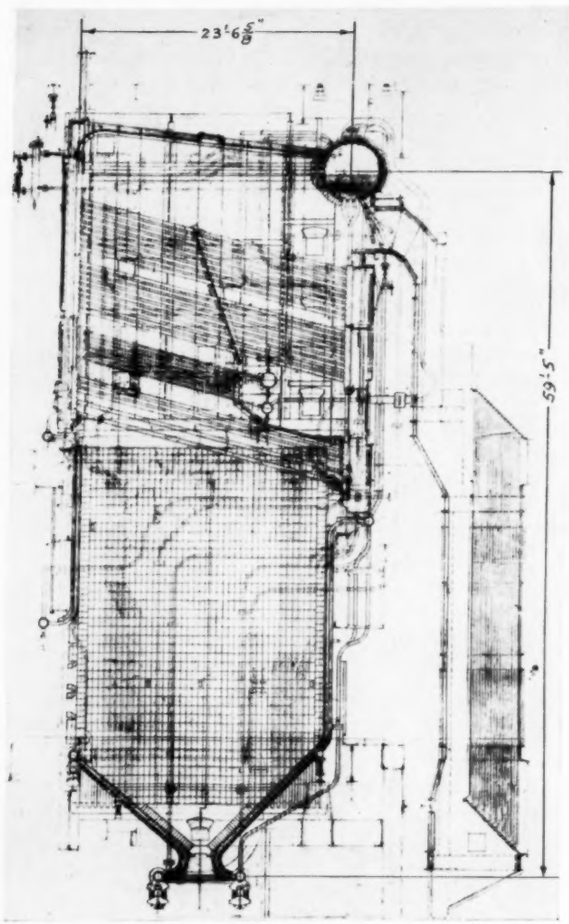

FIG. 22.

cooled hopper-bottom was installed to permit the use of pulverized coal at a later date, if desired. These units built for 450 pounds pressure operate at a capacity of 350,000 to 400,000 pounds an hour burning gas and fuel oil. At 400,000 pounds of steam per hour the heat release is:

B.t.u. per cubic foot	24,500
" " front foot	16,100,000
" " square foot of cold surface exposed to the furnace	226,000

Philo (6) describes the results obtained from this installation and his paper is well worthy of careful consideration.

Figure 23 shows the side elevation of the 1400 pound pressure units lately installed at Station A of The Pacific Gas and Electric Company. These units operate at a capacity of 500,000 pounds of steam an hour giving a heat release of:

B.t.u. per cubic foot	37,700
" " front foot	13,650,000
" " square foot of cold surface exposed to the furnace	377,500

In these furnaces temperatures of approximately 3000 degrees have been obtained with entirely satisfactory results.

From the foregoing it is evident that the types of furnaces developed during the last fifteen years have met with the approval of those responsible for the production of steam with fuels ranging from the 1850 degree fahrenheit fusing-temperature-ash coal

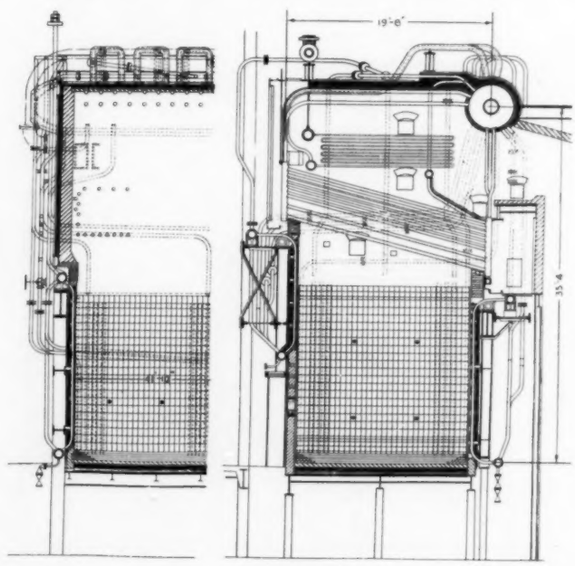

FIG. 23.

(6) In his paper, presented at the Detroit Meeting of the A.S. M. E., June, 1930.

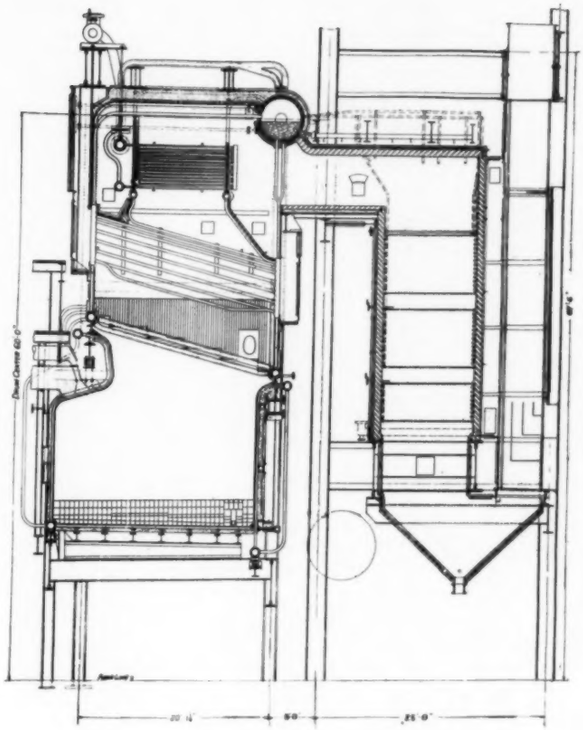

FIG. 24.

of Illinois to the natural gas and fuel oil used on the Pacific Coast.

Within the last two years certain further improvements have been made at State Line in Chicago with the idea of increasing the capacity of furnaces without being limited by the fusing temperature of ash. These experiments, using a poor grade of Illinois coal, have resulted in improved pulverized coal burners, the development of a target floor which would withstand the intense heat produced by the impact of a high capacity flame, and the screening of the boiler from high temperatures of the furnace proper so as to decrease slag difficulties in the boiler.

Figure 24 shows a side elevation of the units now being built for the extension to this plant which include the developments from these experiments. The boilers are of the Two Stage type built for 1400 pound pressure and designed to deliver 500,000 pounds of steam per hour. The furnace has a width of 39 feet. With these burners the fuel and primary air discharge against deflecting castings placed on the arch tubes, join with the secondary air and pass through the spaces between the tubes, thereby projecting the burning fuel and gas downward toward the floor which is covered with molten slag. The floor is completely water-cooled, using Bailey Blocks to prevent the penetration of slag from the high-sulphur coal used at this station. The water-cooled tubes of the furnace are covered with blocks and immediately above the furnace is a dust fluxing row of tubes. Studs are welded by the electric resistance method to the outside of the tubes, giving a porcupine effect. The spaces between the studs are filled with plastic refractory to provide a surface to which the slag can adhere and from which the slag melts and drips down to the floor, where it can be readily tapped out in liquid form. At 500,000 pounds of steam an hour the heat release under the dust fluxing screen will be:

B.t.u. per cubic foot	42,500
" " front foot	15,250,000
" " square foot of cold surface exposed to the furnace	254,000

The importance of these figures will be appreciated when it is realized that the kind of coal burned at this plant has always required considerable cleaning of boilers, even with natural draft stokers, due to the low fusing temperature of the ash.

The increasing capacity of furnaces is well illustrated in Figure 25 in which the line marked A represents the side elevation outline of the low-capacity pulverized-coal furnace of about 1920 which with a 25-foot front had a capacity of 150,000 pounds of steam. The line marked B shows the Calumet boiler at Chicago which generates 250,000 pounds of steam with 25-foot furnace front. The line marked C rep-

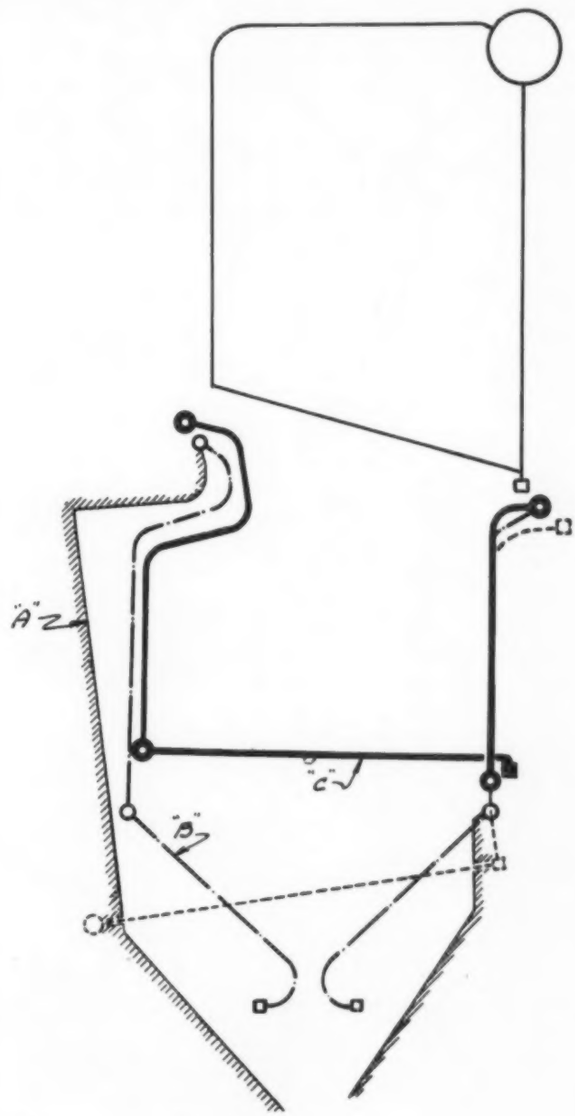

FIG. 25.

	DATE OF INSTALLATION	BOILER	FURNACE					FIRING EQUIPMENT	FUEL		CAPACITY LBS. STEAM/HR.	B.T.U. LIBERATION PER HOUR PER			
			WALLS	FLOOR	WIDTH	DEPTH	HEIGHT		KIND	ASH FUSION TEMP.		FRONT FOOT	CU. FT.	SQ. FT. EXPOSED COOLING SURFACE	SQ. FT. ENVELOPE
1. LEAGUE ISLAND NAVY YARD	1918	B&W EXPRESS TYPE	REFRACTORY	REFRACTORY	11'-4"	10'-3"	6'-10"	B&W MECHANICAL OIL BURNERS	OIL	-	153,467 77,000	10,100,000 4,680,000	291,000 134,000	1,300,000 599,000	330,000 153,000
2. DETROIT EDISON COMPANY DELRAY STATION NO. 2	1910	STIRLING W TYPE	REFRACTORY	STOKER	26'-6"	14'-5"	17'-6"	UNDERFEED STOKERS	W. VA. BITUMINOUS	2700	147,000	4,200,000	27,500	393,000	91,600
3. BUFFALO GENERAL ELECTRIC CO.	1916	B&W	REFRACTORY	STOKER	23'-11"	17'-3"	12'-0"	UNDERFEED STOKERS	PENNA. BITUMINOUS	2200	90,000	2,600,000	35,000	361,000	85,600
4. PUGET SOUND POWER & LIGHT CO. SEATTLE, WASHINGTON	1918	B&W	REFRACTORY	REFRACTORY	6'-3"	11'-6"	15'-0"	PULV. COAL FLARE BURNERS	WASHINGTON BITUMINOUS	2200	13,800	2,470,000	24,100	338,000	34,800
5. TRUMBULL STEEL COMPANY WARREN, OHIO	1922	STIRLING	REFRACTORY	REFRACTORY	15'-0"	11'-0"	14'-0"	PULV. COAL FLARE BURNERS	W. VA. BITUMINOUS	2400	30,200	2,980,000	17,900	177,000	38,100
6. WEST PENN POWER COMPANY SPRINGDALE, PENNA.	1922	B&W	REFRACT.- BARE TUBE REAR	SLAG SCREEN	19'-0"	22'-3"	31'-0"	PULV.COAL STREAM LINE BURNERS	PENNA. BITUMINOUS	2200	180,000	10,250,000	20,600	327,000	72,400
7. UNION ELEC. LIGHT & POWER CO. CAHOKIA STATION	1923	B&W	REFRACT.-BARE TUBE REAR	SLAG SCREEN	17'-0"	23'-0"	31'-3"	PULV.COAL STREAM LINE BURNERS	ILLINOIS COAL	2000	129,000	8,440,000	16,160	222,500	58,600
8. UNION ELEC. LIGHT & POWER CO. CAHOKIA STATION	1925	B&W	BARE TUBE REAR & SIDES	SLAG SCREEN	21'-0"	24'-3"	31'-6"	PULV.COAL STREAM LINE BURNERS	ILLINOIS COAL	2000	140,000	9,260,000	15,400	118,500	56,400
9. UNION ELEC. LIGHT & POWER CO. ASHLEY STREET STATION	1925	EDGEMOOR	SIDES - BLOCK COVERED TUBES	BLOCK COV'D TUBES	11'-9"	19'-0"	18'-6"	PULV. COAL CIRCULAR BURNERS	ILLINOIS COAL	2000	85,000	6,208,000	23,120	133,300	45,700
10. UNION ELEC. LIGHT & POWER CO. CAHOKIA STATION	1927	B&W	SIDES-TUBE & TILE ARCH & REAR - BLOCK COV'D TUBE	BLOCK COV'D HOPPER	19'-2"	20'-8"	33'-0"	PULV. COAL CIRCULAR BURNERS	ILLINOIS COAL	2000	240,000	13,500,000	24,200	184,500	88,600
11. COMMONWEALTH EDISON COMPANY CALUMET STATION	1926	B&W	BLOCK COV'D TUBES	BLOCK COV'D HOPPER	23'-8"	19'-9"	26'-9"	PULV. COAL INTERTUBE BURNERS	KINCAID COAL	1850	250,000	14,700,000	28,200	224,000	108,500
12. BUFFALO GENERAL ELECTRIC CO. HUNTLEY STATION	1926	B&W	BLOCK COV'D TUBES	SLAG TAP	24'-4"	15'-7"	19'-6"	PULV. COAL INTERTUBE BURNERS	PA. MIXED BITUMINOUS	2200	275,000	8,486,000	40,450	314,000	140,000
13. EDISON ELEC. ILLUMINATING CO. EDGAR STATION	1929	B&W	SIDES & REAR-BLOCK COV'D TUBE	STOKER	28'-0"	18'-0"	20'-9"	UNDERFEED STOKER	NEW RIVER COAL	2600	300,000	16,000,000	42,500	316,000	152,000
14. UNITED ELECTRIC LIGHT & POWER CO. NEILGATE STATION	1930	B&W	BLOCK COV'D TUBES	SLAG TAP	34'-4"	41'-0"	30'-0"	PULV. COAL CROSS TUBE BURNERS	POCAHONTAS COAL	2450	800,000	16,750,000	28,200	243,000	152,000
15. JERSEY CENTRAL POWER & LIGHT CO. SOUTH AMBOY, NEW JERSEY	1930	B&W	BLOCK COV'D TUBES	WATER COOLED SLAG TAP	27'-2"	18'-5"	26'-6"	PULV. COAL CROSS TUBE BURNERS	W. PA. BIT. POCAHONTAS	2180 2450	280,000	13,440,000	27,500	207,500	102,500
16. AMERICAN GAS & ELECTRIC CO. NEWARK, NEW JERSEY	1930	B&W	TUBE & TILE AND BLOCK COV'D TUBES	SLAG TAP	34'-11"	17'-11"	31'-0"	PULV. COAL CROSS TUBE BURNERS	CENTRAL PA. BITUMINOUS	2200	330,000	16,130,000	22,600	167,500	96,500
17. DIAMOND ALKALI COMPANY PAINESVILLE, OHIO	1930	B&W	BLOCK COV'D TUBES	SLAG TAP	23'-8"	20'-9"	30'-0"	PULV. COAL INTERTUBE BURNERS	PA. & OHIO BITUMINOUS	2100-2500	250,000	15,206,000	24,370	148,000	97,500
18. SUPER POWER CO. OF ILLINOIS POWERTON STATION	1931	B&W	BLOCK COV'D TUBES	SLAG TAP	36'-0"	18'-10"	31'-3"	PULV. COAL CROSS TUBE BURNERS	ILLINOIS COAL	1950	350,000	12,380,000	22,600	153,500	93,500
19. CHICAGO DIST. ELEC. GEN. CORP. STATE LINE STATION	1929	B&W	BLOCK COV'D TUBES	BLOCK COV'D HOPPER	34'-9"	19'-11"	30'-9"	PULV. COAL INTERTUBE BURNERS	KINCAID COAL	1850	480,000	16,600,000	27,500	193,500	122,900
20. SO. CALIFORNIA EDISON CO. LONG BEACH, CALIF.	1930	B&W	BLOCK COV'D TUBES	TILE & BLOCK COV'D HOPPER	30'-3"	20'-11"	28'-9"	COMBINATION OIL & GAS BURNERS	NATURAL GAS	-	400,000	16,100,000	24,500	226,000	108,000
21. PACIFIC GAS & ELECTRIC CO. STA. "A", SAN FRANCISCO, CALIF.	1931	B&W	BLOCK COV'D TUBES	REFRACTORY	41'-10"	16'-8"	21'-9"	COMBINATION OIL & GAS BURNERS	OIL	-	500,000	13,650,000	37,700	377,500	146,000
22. DETROIT EDISON COMPANY DELRAY STATION NO. 3	1929	STIRLING SPECIAL	BLOCK COV'D AND BARE TUBES	STOKER	26'-3"	22'-7"	21'-5"	UNDERFEED STOKER	W. VA. BITUMINOUS	2300	400,000	20,400,000	39,800	275,000	164,000
23. CHICAGO DIST. ELEC. GEN. CORP. STATE LINE STATION	1932	B&W	STUD & BLOCK COVERED TUBES DUST FLUXING SCREEN	WATER COOLED SLAG TAP	38'-8"	21'-6"	18'-3"	PULV. COAL MULTIPLE INTER-TUBE BURNERS	KINCAID COAL	1850	500,000	15,250,000	42,500	254,000	151,200

TABLE I.

FURNACES—1916-1932

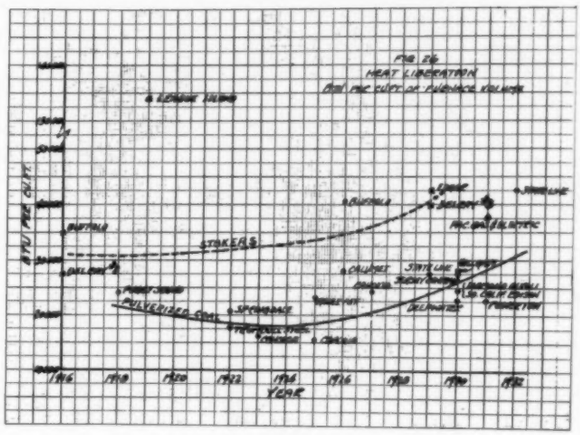

FIG. 26.

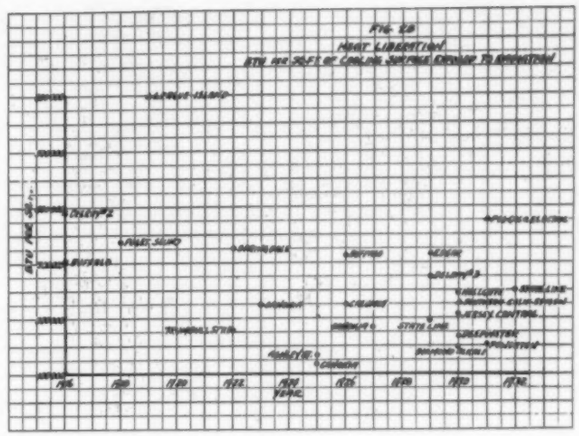

FIG. 28.

resents the new units for Chicago District Electric Generating Corporation, State Line Station, which, based on an equivalent of a 25-foot front furnace, will develop 325,000 pounds of steam per hour. All three arrangements are fired from one side.

Table I lists the important factors of the various furnaces described and shows clearly how the increase in capacity of furnace has been met.

The capacity decreased with the advent of pulverized coal and then gradually increased from about 15,000 B.t.u. per cubic foot to over 40,000 B.t.u.

The B.t.u. per front foot has increased from about 4,000,000 to approximately 16,000,000 or roughly four to one.

The B.t.u. per square foot of cold surface exposed, which is a measure of furnace temperature and in this comparison is an indication of the temperature permissible, is shown from 300,000 to 400,000 B.t.u. per square foot in the stoker installations of 1918 with no cooling of the furnace except by the boiler. Little change is shown in this figure for stoker-fired furnaces, the increase in capacity demanded being met by extending the amount of cold surface from the boiler proper to the walls. With pulverized coal this value seems to be approaching approximately 200,000 B.t.u. per square foot.

The B.t.u. per square foot of envelope has increased from approximately 60,000 on the early pulverized-coal burning units to approximately 110,000 B.t.u. per square foot on the later installations.

These figures are perhaps more clearly shown by the following diagrams:

Figure 26 shows the B.t.u. liberated per cubic foot for stoker-fired and pulverized-coal fired units. A dotted reference line is shown for stokers and a full line for pulverized coal. While these reference lines may not represent true averages because of the small number of points for stoker-fired boilers and because of the scattered results for boilers fired with pulverized coal, they will be of assistance in making a general comparison.

Figure 27 shows the B.t.u. liberated per square

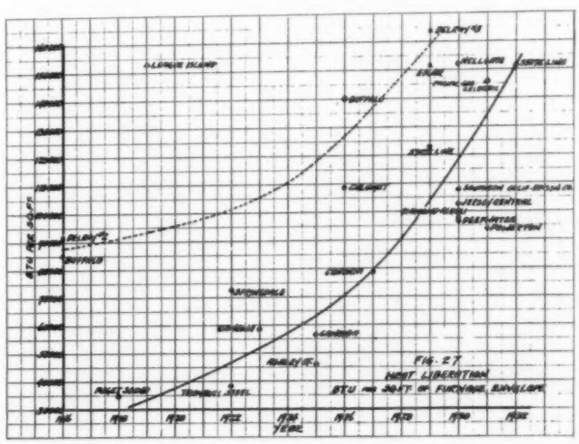

FIG. 27.

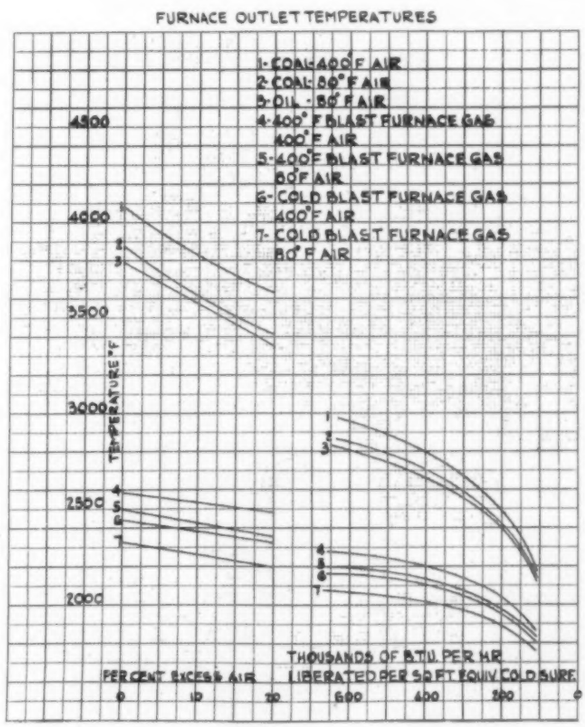

FIG. 29.

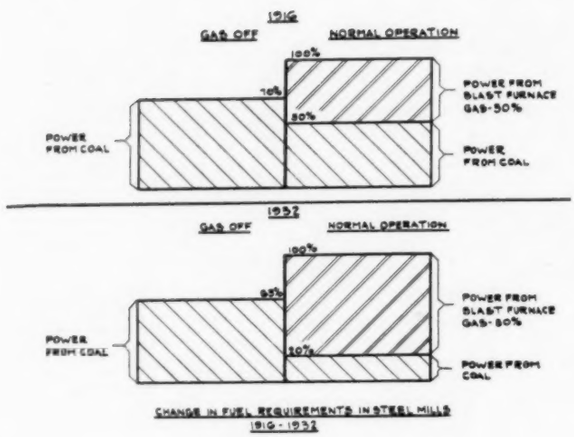

FIG. 30.

foot of envelope for pulverized coal fired and stoker-fired furnaces.

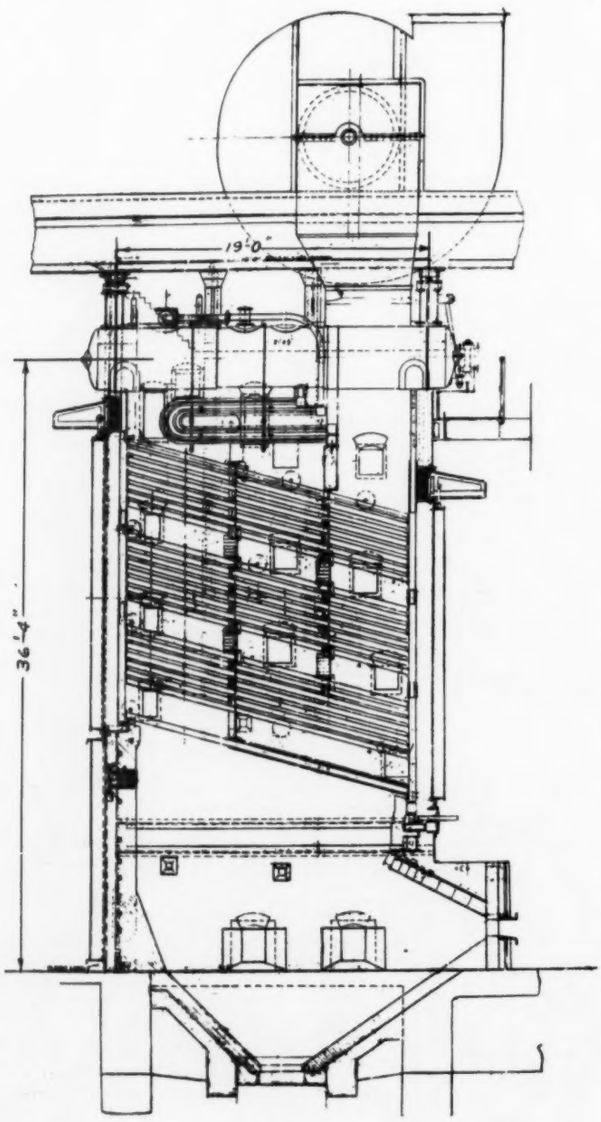

FIG. 31.

Figure 28 shows the B.t.u. liberated per square foot of cold surface. In the case of pulverized coal this figure is about 200,000 B.t.u. per square foot for coals containing ash approximately 2200 degrees fusing temperature. For stoker-fired boilers this value remains about 300,000 B.t.u. per square foot. For oil this figure ranges from the ideal set of about 600,000 down to about 350,000. This reduction appears to be a penalty due to size of unit. All the surface of a small size unit can be considered as cooling the combustion flame, but as size increases this is no longer true, and conditions prevailing at the burner zones of the furnace are materially different from those at the exit.

While the foregoing account of furnace development at central stations has a direct bearing on steel mill power plants, consideration should be given to the differences which exist in both fuel and conditions for the two classes of service.

Figure 29 shows calculated exit furnace temperatures with various fuels including blast furnace gas. On the left side are shown theoretical temperatures with varying percentages of excess air up to 20%. On the right hand side are shown calculated furnace-exit temperatures with heat releases from 600,000 down to 100,000 B.t.u. per square foot of cooling surface exposed to the furnace, assuming 20% excess air.

Starting from the top the curves show:

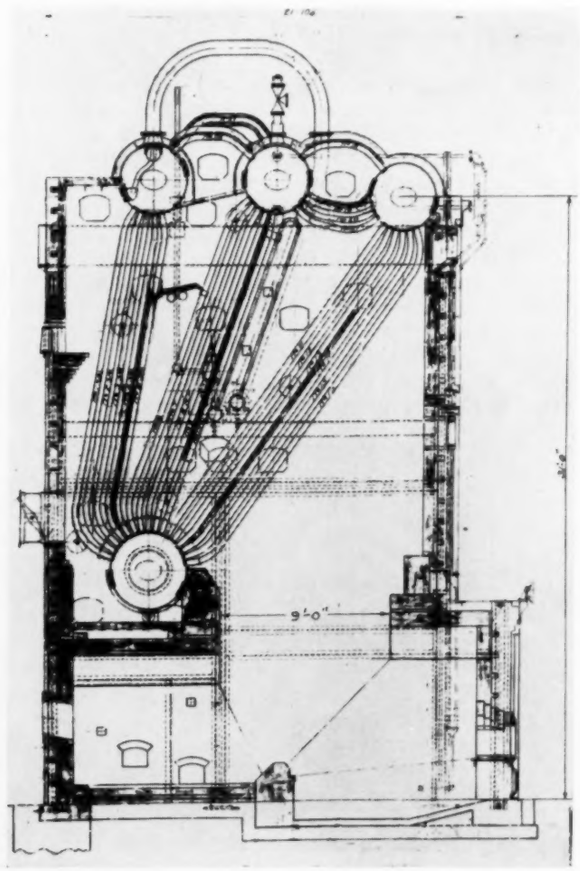

FIG. 32.

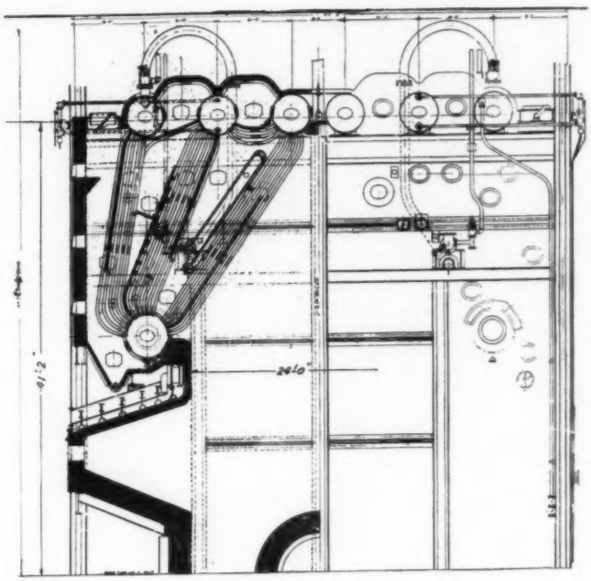

FIG. 33.

1. Good grade of coal with 400 deg. fahr. air.
2. Good grade of coal with cold air.
3. Fuel oil with cold air.

In the lower group:

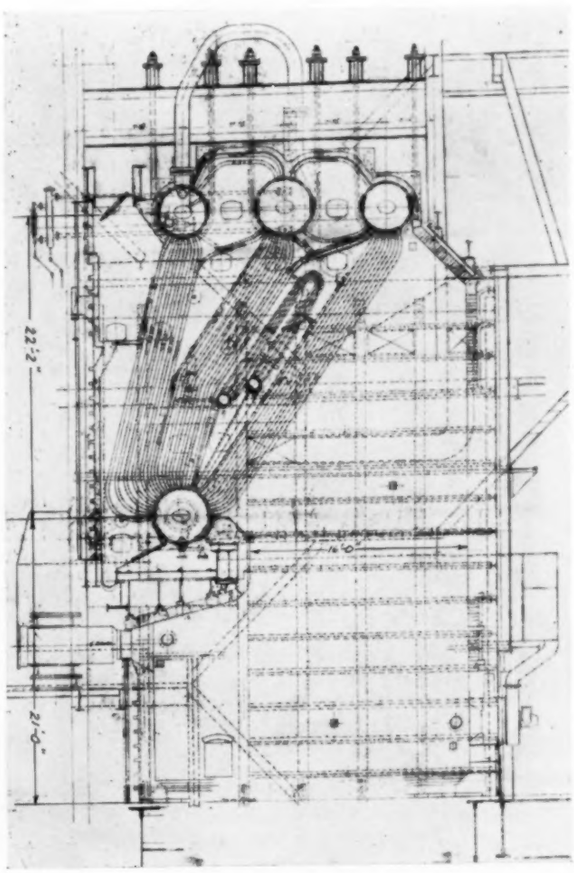

FIG. 34.

4. 400 deg. fahr. blast furnace gas and 400 deg. fahr. air.
5. Cold blast furnace gas and 400 deg. fahr. air.
6. 400 deg. fahr. blast furnace gas and cold air.
7. Cold blast furnace gas and cold air.

The calculations of temperatures on the right hand side of the diagram are made on the assumption that the hottest part of the furnace is 200 degrees higher in temperature than the exit and one-half of the furnace cold-surface is exposed to radiant heat at this temperature. The remaining half of the cold surface is exposed to radiant heat at the outlet temperature. These calculations check with the results of many installations using 50% black body radiation.

The figure shows that with blast furnace gas the limit for furnace-exit temperature is reached at far higher liberation rates per square foot of cold surface than in the case of pulverized coal furnaces. It indicates that a liberation rate of 600,000 B.t.u. per square foot of cold surface should be obtainable in comparison with approximately 200,000 B.t.u. with pulverized coal, even if we assume which may not continue to be necessary that blast furnace gas carries dust of 2300 degrees fusing temperature.

The graphs, taken in conjunction with the pulverized-coal furnaces described, show clearly that no furnace-temperature limits are being approached with blast furnace gas only. The furnace problem today for the steel mill requires attention, largely because conditions have changed during the last 20 years as illustrated in Figure 30.

In the upper half is shown what is termed 1916 conditions. To the right is shown normal operation with 50% of the power from coal, and to the left the gas-off condition, with coal furnishing all the power, the increase in coal burning rate being less than 50% above normal. In those days steel mill power plants consisted of long rows of small boilers and the large steel mills had several of these rows. Some of them were fired by blast furnace gas, usually with a set of grate bars in the furnace to carry a small amount of coal for ignition purposes should the gas go off. Other rows of boilers were fired by stokers of one kind or another with no connection to the blast furnace gas. Therefore, when the gas went off, it was simply necessary to increase the rating somewhat on the stoker-fired units until the gas came back. Under these conditions, furnaces were designed either for blast furnace gas or for coal, but not for both.

In the lower half of the figure are shown what might be termed 1932 conditions. Again, on the right hand side is shown normal operation, and on the left hand side the gas-off conditions. Here the coal rate is shown to increase 300% with the blast furnace going off. The contrast between the upper half and the lower half is quite a credit to the mill steam engineer who is still making improvements and probably will not rest until he has eliminated the use of 20% coal, which has already been lowered at some mills.

The change shown has presented certain problems of furnace design due to the fact that it is now economical to use the same boilers for all conditions using whatever fuel may be required at any time, either singly or in combination.

Figure 31 shows the boilers installed at Plant A

about 1918. These units, 16 feet wide, set over all brick furnaces of 4,240 cubic feet, were fired by burners which mixed the gas and air on entering the furnace. These units operated at 68,000 pounds of steam per hour with an over all efficiency of approximately 76% without air heaters or economizers. The heat release in these furnaces was:

B.t.u. per cubic foot	24,700
" " front foot	6,540,000
" " square foot of cold surface exposed to the furnace	427,000

For the purpose intended, there is little doubt but that these furnaces were quite satisfactory and would last indefinitely.

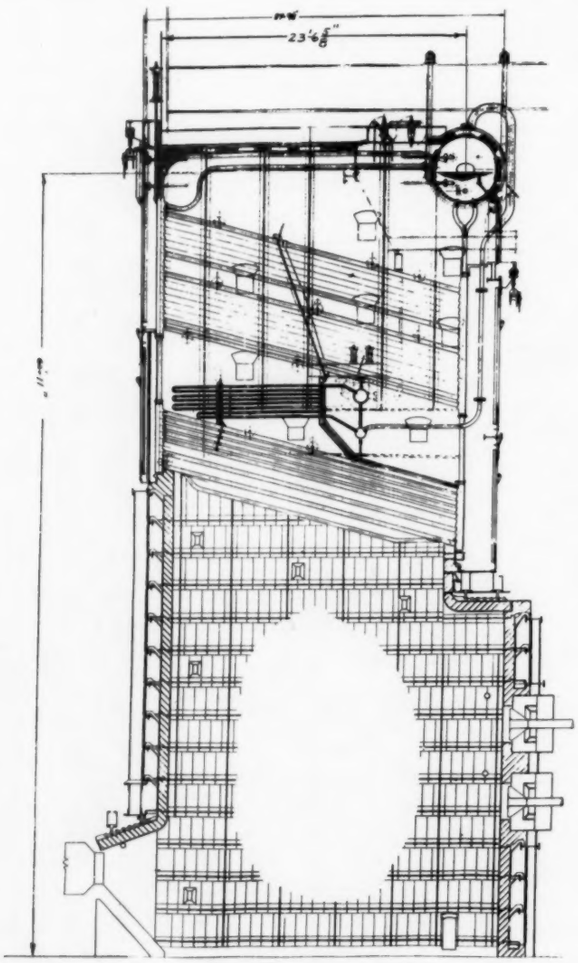

FIG. 35.

Figure 32 shows an installation made about 1920 at Plant B. These units were installed to burn blast furnace gas only. The gas-off condition was taken care of by another power house. Stirling boilers of 795 horsepower were used without preheated air. These units operated at approximately 45,300 pounds of steam an hour or 200% of rating and no cooling surface was exposed to the furnace other than the boiler. The heat release in these furnaces was:

B.t.u. per cubic foot	27,000
" " front foot	5,020,000
" " square foot of cold surface exposed to the furnace	343,000

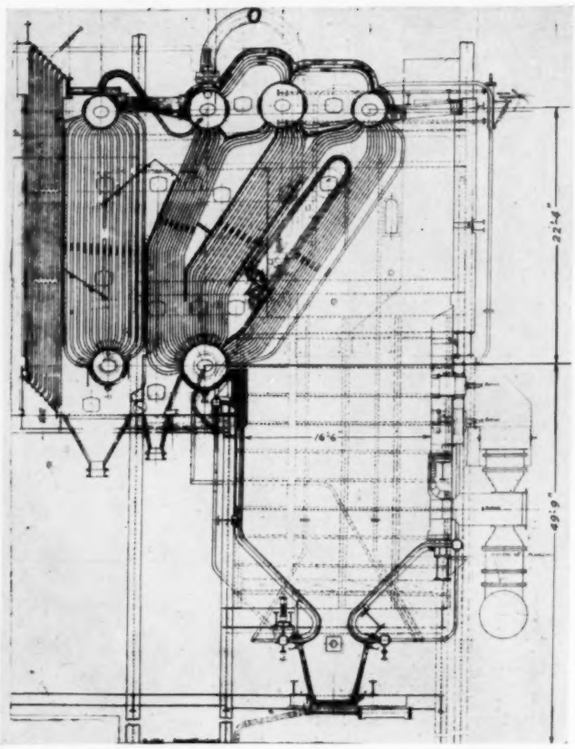

FIG. 36.

A notable installation was made at Plant C about this time, which consisted of eight twin Stirling boilers, four for blast furnace gas alone, three for blast furnace gas and underfeed stokers, and one for coke breeze alone fired by chain grate stokers. Figure 33 shows one of the units operated with blast

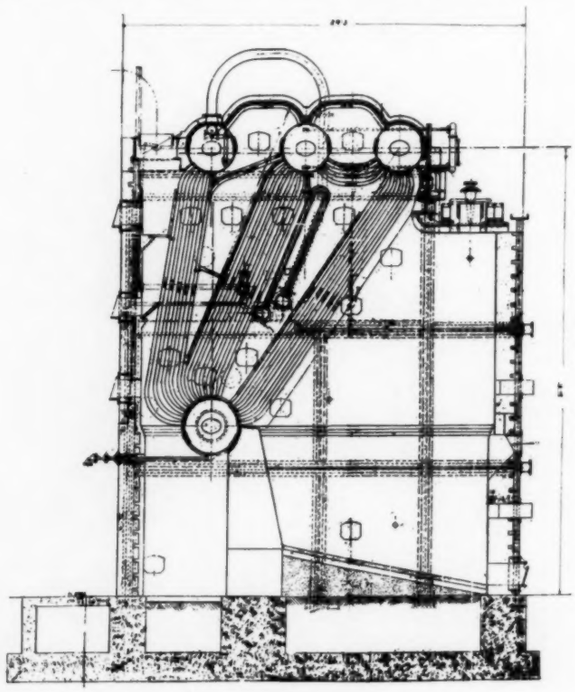

FIG. 37.

furnace gas alone. Using mixing burners, and all-brick furnaces for boilers operating at 200,000 pounds of steam per hour with an efficiency of approximately 75%. The heat release was:

B.t.u. per cubic foot	21,300
" " front foot	8,000,000
" " square foot of cold surface exposed to the furnace	466,000

No trouble whatever has been experienced with these brick furnaces.

Figure 34 shows the unit installed in Plant D in 1928 for 165,000 pounds of steam an hour with blast furnace gas. This unit burns a combination of blast furnace gas and pulverized coal, the burners being located in opposite walls. The blast furnace gas burners are located under the arch at the mud drum.

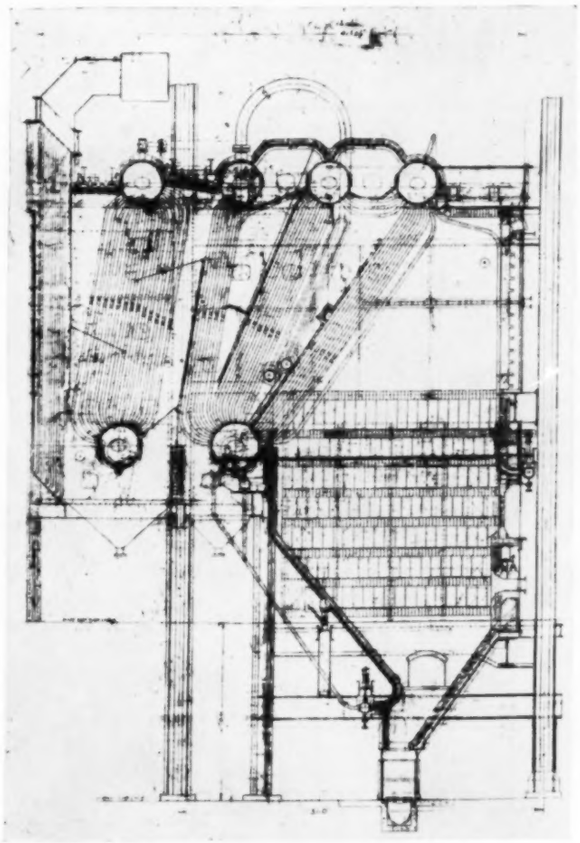

FIG. 38.

The pulverized coal burners are located in the front wall at a lower level.

With blast furnace gas this furnace with all-refractory walls is within the temperature limitations for which it was built and it was undoubtedly intended to be operated primarily with blast furnace gas. If, however, these furnaces were required to handle the full load with coal, it would be necessary to add water cooling to the furnace.

The heat release with blast furnace gas at the capacity given is:

B.t.u. per cubic foot	12,500
" " front foot	8,750,000
" " square foot of cold surface exposed to the furnace	389,000

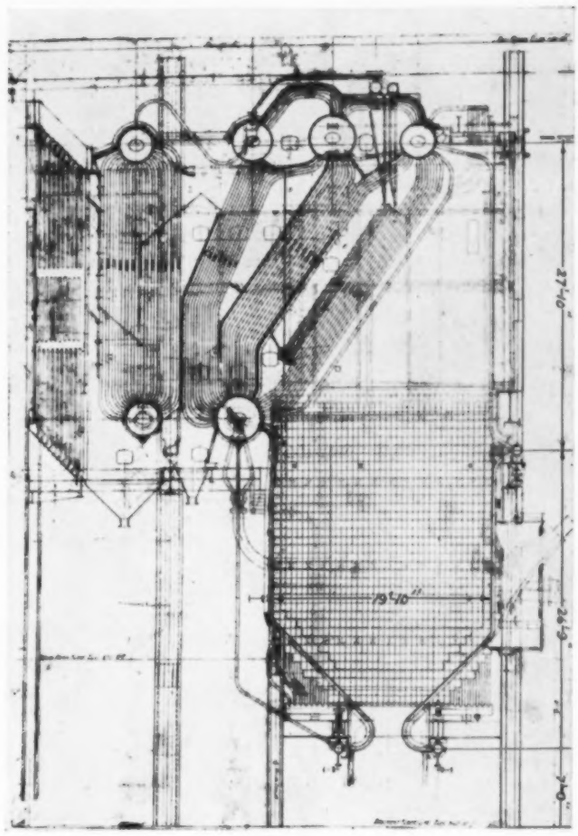

FIG. 39.

Figure 35 shows an installation made at Plant E in 1930. This plant was designed primarily for blast furnace gas but was also equipped to burn pulverized coal as might be demanded. Burners are placed in opposite walls, the pulverized-coal burners being above the elevation of the blast-furnace-gas burners. There is no water cooling in the furnace other than by the boiler, notwithstanding that when operating at 208,000 pounds of steam per hour, the heat release was:

B.t.u. per cubic foot	15,000
" " front foot	13,259,000
" " square foot of cold surface exposed to the furnace	642,000

Figure 36 shows an installation at Plant F which

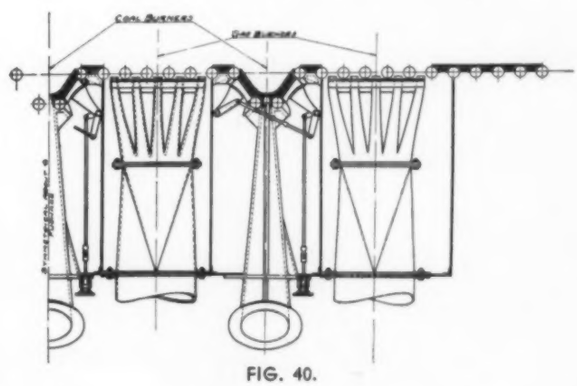

FIG. 40.

	DATE OF INSTALLATION	BOILER	FURNACE				FIRING EQUIPMENT	FUEL		CAPACITY LBS. STEAM/HR.	B.T.U. LIBERATION PER HOUR PER			
			TYPE	WIDTH	DEPTH	HEIGHT		KIND	ASH FUSION TEMP.		FRONT FOOT	CU.FT.	SQ.FT. EXPOSED COOLING SURFACE	SQ.FT. ENVELOPE
PLANT "A"	1919	B&W	REFRACTORY	16'-0"	17'-0"	14'-9"	MIXING BURNER	BLAST FURNACE GAS - WASHED	-	68,000	6,540,000	24,700	427,000	70,300
PLANT "B"	1921	STIRLING	REFRACTORY	14'-6"	9'-0'	16'-0"	MIXING BURNER	BLAST FURNACE GAS - WASHED	-	45,300	5,020,000	27,000	343,000	37,900
PLANT "C"	1924	STIRLING TWIN	REFRACTORY	20'-0"	30'-8"	22'-9"	MIXING BURNER	BLAST FURNACE GAS - DIRTY	-	200,000	8,000,000	21,300	466,000	78,600
PLANT "D"	1928	STIRLING	REFRACTORY AIR COOLED	30'-0"	23'-0"	28'-6"	B. F. GAS-MIXING PULV. COAL-CIRCULAR	B.F.GAS-.01 GRS./CU.FT. BITUMINOUS	2200	165,000	6,750,000	12,500	389,000	78,000
PLANT "E"	1930	B&W	REFRACTORY AIR COOLED	23'-8"	26'-0"	35'-0"	B. F. GAS-MIXING PULV. COAL-CIRCULAR	B.F.GAS-.75 GRS./CU.FT. PENNA. BITUMINOUS	- 2100	208,000	13,259,000	15,000	642,000	68,200
PLANT "F"	1931	STIRLING	REAR WALL & HOPPER BOTTOM-BLOCK COV'D TUBES; SIDES-AIR COOLED REFRACTORY	19'-0"	16'-6"	25'-0"	B.F. GAS - MIXING TAR -	B.F.GAS-.01 GRS./CU.FT.	-	125,000	9,380,000	20,350	282,000	65,900
PLANT "G"	1925	STIRLING	REFRACTORY	14'-6"	16'-9"	16'-0"	B.F.GAS-ASPIRATING P.C.-FLARE BURNER	BLAST FURNACE GAS - DIRTY BITUMINOUS	- 2000-2200	50,000 50,000	4,500,000 4,500,000	14,200 14,200	267,000 267,000	38,900 38,900
PLANT "H"	1928	STIRLING	REAR WALL & HOPPER BOTTOM-BLOCK COV'D TUBES; FRONT HOPPER-REFRACTORY; FRONT WALL-BARE TUBE SIDES-AIR COOLED	22'-8"	19'-0"	21'-6"	B.F.GAS-ASPIRATING PULV. COAL-CIRCULAR	B.F.GAS-.1 GRS./CU.FT. BITUMINOUS	- 2000-2200	175,300 131,500	10,900,000 7,950,000	20,600 14,800	286,000 207,000	76,800 55,600
PLANT "I"	1932	STIRLING	BLOCK COV'D & BARE TUBES; BLOCK COV'D HOPPER	23'-2"	19'-10"	30'-6"	B. F. GAS-HORIZONTAL INTERTUBE PULV. COAL-CALUMET	BLAST FURNACE GAS BITUMINOUS	- 2000-2200	200,000 300,000	12,000,000 17,300,000	16,870 24,250	151,000 217,000	69,200 99,300

TABLE 2.

Blast Furnace Gas Furnaces—1918-1932

was originally equipped to burn blast furnace gas and fuel oil, possibly to be changed at some later date to pulverized coal. The rear wall was made of block-covered tubes with hopper bottom. The side and front walls were refractory lined, the side walls being air cooled.

At 125,000 pounds of steam per hour, burning blast furnace gas the heat release was:

B.t.u. per cubic foot	20,350
" " front foot	9,380,000
" " square foot of cold surface exposed to the furnace	282,000

All burners are located in the front wall the fuel oil burners being above the blast-furnace-gas burners.

Perhaps the development of steel mill Boiler furnaces from the advent of combination firing of pulverized coal and blast furnace gas can be most readily seen by comparing installations made at the same plant over a period of years. We find this development exemplified at Plant G.

Figure 37 shows an installation made about ten years ago, which combined pulverized coal and blast furnace gas without any cold surface in the furnace.

These units operated satisfactorily at approximately 200% of rating burning dirty hot blast furnace gas with cold air. At this rating, which correspond to about 50,000 pounds of steam an hour, the heat release was:

B.t.u. per cubic foot	14,200
" " front foot	4,500,000
" " square foot of cold surface exposed to the furnace	267,000

The next installation for this same company, made in 1928, is shown by Figure 38 which shows a 46 wide Stirling boiler of 1460 horsepower, designed for a working pressure of 375 pounds. The furnace cooling consisted of air-cooled side walls block covered hopper bottoms and bare tubes protecting the front wall. These units were fired with both pulverized coal and blast furnace gas at capacities up to 400%, which corresponds to about 175,300 pounds of steam per hour. The over-all efficiency of the units under these conditions was about 83%, the draft loss through the units being 7.7".

Hot clean blast furnace gas, containing .1 grain of dust per cubic foot, was the major fuel supply and, when burned with hot air at the capacity stated, the heat release in the furnace was:

B.t.u. per cubic foot	20,600
" " front foot	10,900,000
" " square foot of cold surface exposed to the furnace	192,000

Burning coal at a rate to produce 131,500 pounds of steam an hour the heat release was:

B.t.u. per cubic foot	14,800
" " front foot	7,950,000
" " square foot of cold surface exposed to the furnace	139,000

The next installation at this plant, which is about ready for operation, is shown in Figure 39. This installation consists of a 46 wide Stirling boiler of 2100 horsepower with all-welded drum construction fitted with economizers and air heater. As shown, the furnace consists of block covered walls in the lower furnace, the upper front wall being protected by a row of bare tubes, and the upper side walls of brick. An interesting development is shown in the location of the pulverized coal and gas burners which are all located on the front wall on one horizontal line. The burners consist of three Intertube type pulverized coal burners and four blast furnace gas burners. These units are designed to operate at 200,000 pounds of steam an hour with all-blast furnace gas, or with 50% blast furnace gas and 50% coal. With 100% pulverized coal the units are designed to operate at 300,000 pounds of steam an hour.

Figure 40 represents a section through the front wall showing the burner arrangement.

Under the blast furnace gas conditions the efficiency will be 85% and the total draft loss 6.8"; the heat release will be:

B.t.u. per cubic foot	16,870
" " front foot	12,000,000
" " square foot of cold surface exposed to the furnace	117,500

At 300,000 pounds of steam an hour burning pulverized coal, the over-all efficiency will be somewhat better than 86% with a draft loss of 6.7" giving a heat release of:

B.t.u. per cubic foot	24,250
" " front foot	17,300,000
" " square foot of cold surface exposed to the furnace	169,500

These units represent a furnace design capable of handling pulverized coal at a capacity approximating that common to large central station power house boilers.

Figure 41 shows a table of factors of furnaces for blast furnace gas similar to the one shown for pulverized coal installations. This table brings out the variations which have taken place and how these have followed the advent of pulverized coal applied to the conditions of steel mills. These figures show the tendency of blast furnace gas boiler furnaces to be approaching the outlet temperature limitations set by pulverized coal.

The combination of fuels also brings in the problem of location and protection of burners for both fuels. From the standpoint of combustion conditions it seems desirable to place all burners in a single line, thus permitting of maximum flame length for either fuel with as little interference as possible by flame from one fuel with the flame from the other. Also, since only one fuel may be in use alone at a given time, it appears highly desirable to have the burners protected from the furnace heat by water-cooling tubes. The method now in more general use of maintaining a small amount of air flow through the burners not supplying fuel is obviously not desirable, though it has been found to be necessary in order to cut down the maintenance, particularly on pulverized coal burners.

To sum up—theoretical calculations show and furnaces in successful operation prove:

1. With blast furnace gas fuel only, no water cooling other than that by the boiler is required in furnaces to meet present-day demands.
2. With the improvements which have been made in steel mill power generation, the use of pulverized coal in the same furnaces as the blast furnace gas is required. As a result, boiler furnaces will require water cooling when the conditions reach the limitations found to exist in this fuel.

The practical limitations are:

The intense combustion-zone temperature-effect on the walls—no brickwork can withstand present-day rates with low fusing temperature-ash coals.

The removal of ash which is above the fusion temperature. Either a water-cooled or a liquid-ash bottom is necessary.

Reduced furnace-end temperature. This must be accomplished to a degree that will eliminate serious slagging of boilers. Examination of the various furnaces shown indicates that sufficient cold surface is required so that the liberation rate will not exceed 200,000 B.t.u. per square foot of cold surface, when the ash fusing temperature is 2200 deg. fahr.

1 and 2 may be considered the two extremes met with in steel mill practice. Between these extremes are intermediate conditions starting with mostly blast furnace gas and little coal and going to the conditions represented by the last figure, where the furnaces are required to withstand 30% more heat liberation with all pulverized coal than with blast furnace gas.

Therefore, the amount of water cooling required, if any, will depend upon the heat liberation with pulverized coal. The method of calculation shown by Figure 29 permits of the determination of the amount necessary.

▲

Lubricants for Anti-Friction Bearings

By O. L. MAAG
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Presented before Chicago and Pittsburgh District Sections of the Association of Iron and Steel Electrical Engineers, Chicago, March, 1932; Pittsburgh, April, 1932.

In the lubrication of anti-friction bearings, we use the same lubricants that are used on other types of bearings: namely, mineral oils of various degrees of refinement ranging from light bodied, so-called, spindle or neutral oils to heavy bodied cylinder stocks; greases made to various consistencies compounded from various soaps such as lime, soda, aluminum, etc.; also, compounded oils and greases, such as lead soap, sulfur and chlorinated products to take care of certain conditions which must be met in the various applications in which anti-friction bearings are used.

The developments that have been made in alloy steels and their heat treatment, in the design of machinery, and the industrial demands for more production with the resulting higher speeds and heavier unit loads have placed on the bearings new demands. These can be met only by paying proper attention to the lubricants that are used on these newer units.

The usual physical tests made on oils from which we must govern their suitability for applications are as follows:

Be.
Flash
Fire
Viscosity
Color
Cold Test
Emulsion Test
Gasoline Test
Carbon Residue
Corrosion and Discoloration
Acidity.

To the above tests on oils two others should now be added:

Load Carrying Ability
Abrasion.

For determining the above named properties, the Timken Roller Bearing Company developed a piece of apparatus which has proven very profitable for comparing different lubricants for these two properties.

Fig. 1 shows the cross-section and end view of this apparatus. A mandrel is mounted on two Timken bearings held in a cast-iron base and having on one end of the mandrel a tapered end to receive a standard Timken bearing cup, which is used as one of the test members. A nut having a left-hand thread securely locks this tapered bore cup to the mandrel, with no possible tendency for it to rotate. Directly beneath the revolving cup a test block ½ in. square by ¾ in. long is mounted in a lever, shown in the end view of the apparatus. This lever is supported on a knife edge which is carried on the top of a second lever, which also is carried on a universal knife edge on the base of the machine. This arrangement assures that the test block will at all times be parallel to the revolving cup, so that the unit loading will be maintained constant over the length of the test piece. The top lever is loaded by means of weights on the end, which produces a pressure on the revolving member ten times the amount of load added to the lever.

TEST PROCEDURE

The fluid lubricant to be tested is placed in the 1-gal. tank mounted on the top of the base casting, and brought to the required temperature by means of the electric heating unit directly beneath the oil tank. The oil flows by gravity from the tank over the test piece, the flow being controlled by a valve, shown in the illustration. The arbor is then driven at any desired speed depending on the conditions to be met, either by means of a belt or by being coupled directly to an electric motor. The oil is returned to the tank by a pump inside the base of the machine and driven from the mandrel by a small chain belt, as shown in the diagram. The machine is therefore

a self-contained assembly. The test pieces can be any desired material; however, we have standardized for lubricants on test pieces made from carburized steel, case-hardened and ground, the standard cups and blocks having a Rockwell hardness of 60 on the "C" scale. (Fig. 2.)

The machine, as run for checking lubricants uses a square test block which gives a very high initial loading. The test is comparative as the initial areas are the same. The abrasives in the lubricant, however, will change the ultimate area of contact during test depending on the severity of the abrasives. It is more convenient and cheaper to prepare specimens having flat faces than to prepare them having two curved surfaces in contact. If, however, it is desired to maintain a constant unit pressure, then test pieces can be made having curved surfaces to correspond with the round test cup to be used as the stationary test member.

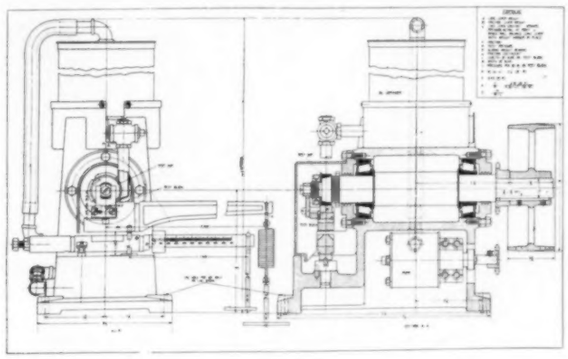

FIG. 1

For testing greases on the machine for film strength, the grease is fed by a pressure gun to the test parts. The grease must be fed at a constant, uniform rate over the usual ten-minute test period. Due, we feel, to the fact that you do not obtain as much cooling with grease as with oil flowing over the test parts, a grease made from a given oil will not show the same load carrying ability or film strength as the oil free from the soap would show; however, there is a definite relation between differently compounded greases.

To determine the comparative abrasive properties of fluid lubricants, we run the test in the same manner as we do for determining its load carrying capacity with the exception that the cups and blocks are weighed before and after a test run on a given lubricant, the loss in weight being a measure of abrasiveness of the lubricant. For extreme pressure lubricants, we made our test runs with a 33-pound lever load (approximately 20,000 pounds per square inch) over a period of six hours, as we feel that a lubricant to be in the extreme pressure class, should carry the above load over a period of time satisfactorily and not permit scoring of metal contacting parts. As quite high losses in the weight of test pieces were obtained in six hours with a number of standard lubricants marketed, it was felt that longer runs were not necessary for comparative data. For the lighter, 10-pound load tests a longer period was deemed desirable, and as a sixteen hour period worked in handily with laboratory operations this period has been used.

The question will naturally be asked, what is there in lubricants that gives them film strength, and what is present that will cause them to be abrasive. In answer to the first question, we submit some tests on oils which show that as the viscosity of a straight mineral oil increases so does its film strength, also, the film strength may be increased by compounding. (Fig. 3).

You will note from the **Be. gravities** of these oils

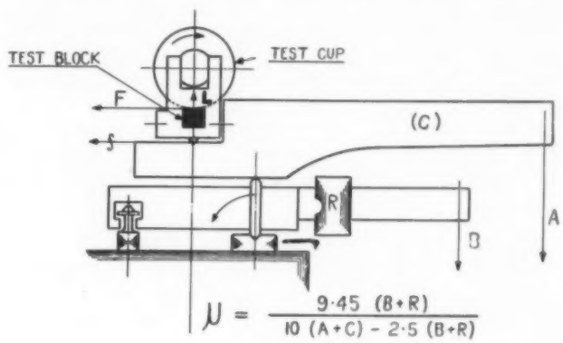

FIG. 2

that both Midcontinent and paraffin base oils are listed, also doped or compounded products. Notice that with the same viscosity there is a very slight advantage in favor of paraffin base oils over the Midcontinent, Sample oils No. 4 and No. 9; however, for practical service, except in isolated cases, this would likely not be sufficient to show up. This difference may be accounted for by differences in the viscosity curve. Usually in cases where extreme pressure lubricants are needed, the unit loads and speeds are such that a much larger load factor must be had than can be obtained by small variations in viscosity of the lubricant. Note the difference that can be obtained by compounding, in oils 13, 14, and 16, whereby with similar viscosities you have a factor of over 2 and 3 to 1 over straight mineral oils. To explain why these two oils show up better in load carrying ability than the straight mineral oils, we wish to state that these particular products contain sulfur and chlorine in a proper form. At present these two elements seem to be the best to be properly compounded with lubricants so that they will have the higher load carrying properties, although marked advantages in film strength can be produced by blowing or oxidizing certain oils; for example, blown rape oil. We have had indications that oxidation of mineral oils increases this property. As in most cases where there are advantages to be had there are difficulties to be overcome. In the use of the above elements, the greatest difficulties seem to come from the corrosive and abrasive angles; however, such rapid strides have been made lately by the oil companies that it appears they will soon have these problems under control. The abrasiveness of lubricants caused by the elements chlorine and sulfur can be reduced to a minimum by proper compounding, as is demonstrated by data obtained on Lubricant No. 6. (Figs. 4 and 5).

Abrasive properties produced by inorganic fillers can likely only be overcome by omitting these substances from the lubricants.

Oil No.	Be. at 60°F.	Flash Deg. F.	Fire Deg. F.	Viscosity — In Sec. 210°F.	130°F.	100°F.	Cold Test Pour Deg. F.	Load OK	Score
1	21.0	325	360			175	0	1#	2#
2	21.0	325	360			301	0	6#	7#
3	22.0	340	390	65		650	+25	7#	8#
4	22.0	345	390	71	275		+35	8#	10#
5	22.7	475	560	133	780		+50	14#	16#
6	22.6	530	620	180	1140		+45	18#	20#
7	20.0	480	570	155			+30	13#	15#
8	27.5	340	385	64	149	296	0	7#	9#
9	26.9	360	410	73	278		+ 5	9#	11#
10	26.6	385	445	99	458		+20	11#	13#
11	25.8	440	510	124			+35	13#	15#
12	25.7	520	600	153			+40	15#	17#
13	21.0	300	350	92	450		0	29#	31#
14	21.3	300	350	84	407		0	15#	18#
15	17.6	460	525	101		2050	0	10#	12#
16	20.6	445	510	76		1100	0	18#	20#

FIG. 3

Data on comparative abrasive properties of gear and bearing lubricants. Loss in weight of test blocks and cups show comparative lapping properties of the lubricants. Data obtained with a rubbing speed of 400 F.P.M. (Fig. 6).

The data in the table indicates that as long as we use clean mineral oils or greases made from clean soaps and oils there is no need to fear from the abrasive properties of lubricants. It also shows the effect of free inorganic fillers when combined with oils.

While the information is not yet conclusive, it appears that the farther away the operating load is from the scoring load of the lubricant, especially in extreme pressure products, the less wear there will be.

Of interest to many will be the results obtained with Dichlorethylether $(CH_2Cl\ CH_2)_2O$ which has a boiling point of 180°C. Although there will undoubtedly be deterioration of film strengths of lubricants using this product due to evaporation, this fault can be remedied somewhat by using soaps in compounding the lubricant. The data indicates that by chlorinating similar higher molar weight compounds, the objection of evaporation can be reduced to a minimum, and the lubricant will still retain the advantages of high film strength and comparatively low abrasive properties. The effectiveness of these film strength giving products is governed, apparently to some extent, by the stock oils, their refinement, etc., so that a given percentage of compound gives varying results when compounded with differently refined stock.

To illustrate how the higher speeds and higher unit loads on bearings both plain and anti-friction being installed today, have affected the lubrication of bearings, we wish to show what the effect of speed and loads has on the film strength of the lubricants. (Fig. 7).

As yet the cases where extreme pressure lubricants are needed on anti-friction bearings are very limited, however, in many instances the same lubricant is used on both gears and roller bearings, so that it is of great importance to be able to have a lubricant that will function properly on both. Very often in gear units in steel mills, heavy sticky lubricants, which are not at all satisfactory for anti-friction bearings, are used in order to protect the gear teeth. By using a compounded lubricant, it is possible to use one of much lighter body and of low cold test, which would serve better on the gears the year round, and also be satisfactory for the anti-friction bearings. A similar product should be satisfactory and very desirable for a year-round lubricant for railway applications.

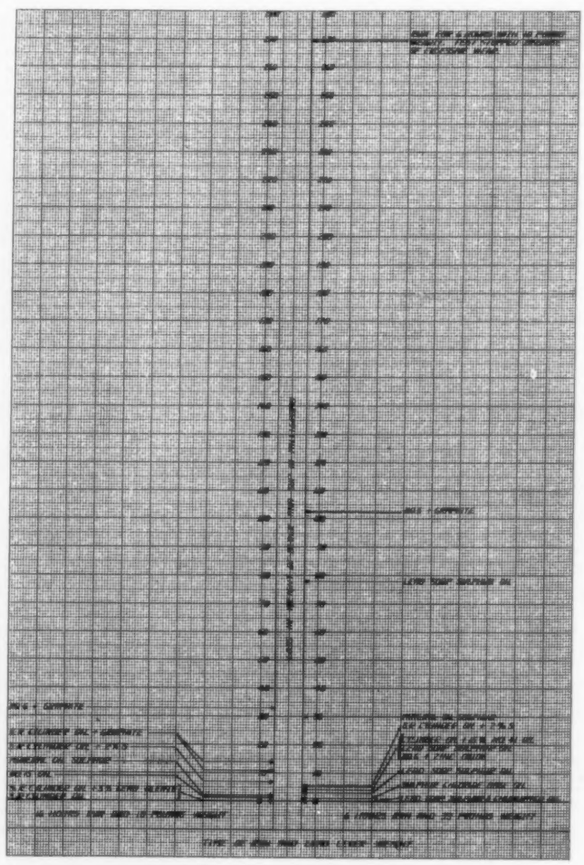

FIG. 4

The sketch shown (Fig. 8) illustrates how both gears and bearings are lubricated satisfactorily by a single lubricant. The large gear dips several inches into the oil so that on revolving it throws lubricant

Lubricant		Time of Test Run	Load Lever Weight	Loss in Wt. of Block & Cup	OK Load
S. R. Cylinder Oil	No. 21	8 hrs.	10#	.0000 gms.	14#
S. R. Cylinder Oil	19	16 "	10#	.0000 "	14#
S. R. Cylinder Oil + 2% S.	2	16 "	10#	.0070 "	40#
S. R. Cylinder Oil + 2% S.	3	6 "	33#	.0055 "	40#
S. R. Cylinder Oil + Graphite 10% D.	20	16 "	10#	.0110 "	14#
S. R. Cylinder Oil + Mica 10%	1	6 "	10#	.2695 "	14#
S. R. Cylinder Oil + Graphite 5% A.	14	16 "	10#	.0140 "	14#
Lead Soap, Sulfur and Chlorinated Oil	6	6 "	33#	.0000 "	85#
No. 6 + Zinc Oxide 10%	7	6 "	33#	.0040 "	85#
No. 6 + Graphite 10% A.	11	16 "	10#	.0330 "	85#
No. 6 + Graphite 10% A.	12	6 "	33#	.1225 "	85#
Lead Soap Sulfur Oil	9	6 "	33#	.0030 "	60#
Lead Soap Sulfur Oil	4	6 "	33#	.0040 "	45#
Lead Soap Sulfur Oil	15	6 "	33#	.0780 "	43#
No. 15 Oil	16	16 "	10#	.0015 "	43#
Mineral Oil + Sulfur	17	6 "	33#	.0300 "	35#
Same as No. 17	18	16 "	10#	.0020 "	35#
S. R. Cylinder Oil + 5% Lead Oleate	31	16 "	10#	.0000 "	16#
Sulfur Chloride Base Oil	41	6 "	33#	.0010 "	85#
Cylinder Oil + 15% No. 41 Oil	42	6 "	33#	.0040 "	43#
Cylinder Oil + 10% Dichlorethylether		6 "	33#	.0020 "	95#
Cylinder Oil + 10% Halowax		6 "	33#	.0065 "	45#

FIG. 5

up on the catch pan and from there it runs by gravity to the roller bearings and thence back to the oil reservoir. A steam coil is installed in the oil reservoir to warm the lubricant in extreme cold weather before starting the unit.

FIG. 6

This system has, in our experience, proven quite satisfactory, and by periodic checking to see that the fluid lubricant is held at the proper level you are always sure to have ample lubrication of both gears and bearings.

Another method that has proven quite satisfactory for the lubrication of anti-friction bearings is to have the oil reservoir so designed that as the bearings revolve the lower part of the rolls dip into the oil. This plan has proven its value for railroad type applications. (Fig. 9).

Note the oil level carried in these units. This level is maintained by periodic inspections and the exact level of oil measured by inserting through the oil plug a graduated gauge.

Other quite common methods used to feed oil to anti-friction bearings are the sight feed bottles, feeding so many drops per minute; also constant level bottle oilers, which are quite satisfactory in many installations such as motors.

The circulating oil systems with filters attached have come into use and have their advantages in many instances. Advantages claimed for them being that the bearings are furnished with clean, cool oil at all times, thus serving to cool the bearings, also to wash out any foreign matter that might get in them. Naturally there is no objection by anti-friction bearing manufacturers to these units.

The oil used in one of these systems must first have the right body or viscosity and properties to protect the gears and bearings. Usually for one of these units an oil ranging in viscosity from approximately 60 sec. at 210°F. to 150 sec. at 210°F. is used, the viscosity varying according to the temperatures and loads encountered.

A well-refined, stable, mineral oil should be used, as an oil that is not somewhat resistant to oxidation will, under the temperatures usually encountered and the constant circulation, oxidize and break down forming products that are acid in nature which form sludge that in turn means oil loss when passed through the filters, or the sludge may even become heavy enough to stop up the oil lines causing bearing trouble.

LUBRICATING GREASES

Many applications of anti-friction bearings for various reasons must be designed to be lubricated with grease. By grease, we mean oils that have been thickened by various soaps, usually lime, soda or aluminum.

The physical properties of greases usually noted are:

Color
Consistency

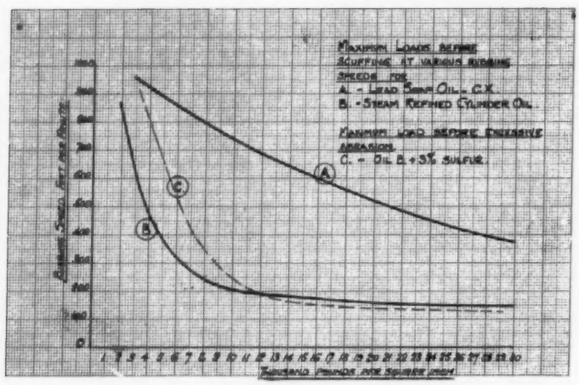
Maximum Loads before Scuffing at various rubbing speeds for
A. - Lead Soap Oil - C.K.
B. - Steam Refined Cylinder Oil.
Maximum load before excessive abrasion
C. - Oil B. + 3% sulfur.
A
B
C

FIG. 7.

Ash
Corrosion
Free Acid or Alkali
Stability (from oil separation on standing)
Type of Soap
Fillers and types
Mineral Oil and its physical properties.

To the above we feel should be added:

Abrasiveness
Load Carrying Ability.

Undoubtedly there are more lubrication troubles traceable to greases on the market than to oils. A

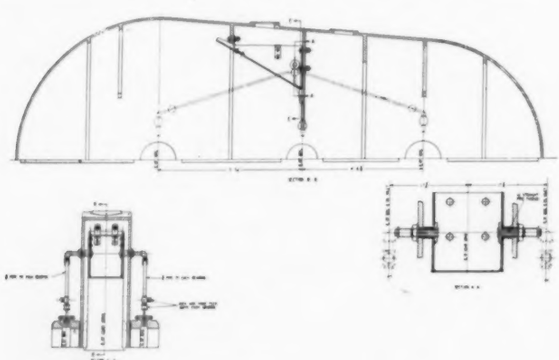

FIG. 8.

buyer of lubricants usually judges a product so much by appearance, and he is easily misled by the appearance of grease lubricants. Also, greases are much more difficult to analyze and to judge their value from the analysis. A further cause is that the grease maker has a real job on his hands to control the consistency of his product, and to keep it from changing its consistency after it has left his plant. Storage conditions, heat, cold and dampness also affect greases much more than oils. It is also easier to use an incorrectly compounded grease than the wrong bodied oil. For example, on a heavy duty unit which requires a good bodied oil for proper bearing protection a grease compounded from a soap and a light bodied neutral oil might be applied since as far as the appearance goes the grease might look even better than the oil. We do not feel we have been far off in asking that greases for different applications be compounded from oils, which would be used on the bearings were the application designed for oil lubrication. We are inclined to feel that more grease troubles can be traced to improper consistency, its variation and change in storage and operation than any other physical property of the grease.

With oils it is difficult to use inorganic fillers such as talc, mica, graphite, asbestos, etc., as they will settle out on standing, also will clog up oil lines, etc. With grease it is not an uncommon practice to incorporate various percentages of these inorganic materials. The abrasive properties of these substances have already been illustrated.

As bearings that must be lubricated with greases operate under all kinds of conditions, such as high and low speeds, light and heavy loads, heat and cold, dry and wet surroundings, and various combinations, it is obvious that each installation must receive proper attention, and the best type of grease recommended to meet the operating conditions. In general, we might say, however, that for heat and high speed conditions, soda soap greases of the proper consistency have proven the most satisfactory. For applications where there is a moisture condition, lime and aluminum soap greases are best. For heavily loaded applications naturally the heavy bodied oils or extreme pressure products are required, and for cold conditions, greases made with low cold test oils should be more satisfactory.

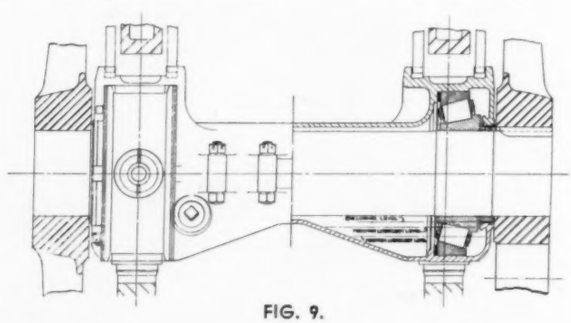

FIG. 9.

For applying greases there are a number of pressure systems which have been worked out that are very satisfactory not only for anti-friction bearings, but also for plain bearing units. The advantages of these systems are that you are more certain of obtaining clean lubricant, also the bearings will receive a definitely determined amount of lubricant at regular intervals.

Conclusion

Modern machinery demands have brought about the use of anti-friction bearings.

Installations should be thoroughly studied by the engineers and designed not only for a proper lubricant but adequate provision should also be made to properly handle the lubricant chosen.

Anti-friction bearings must be properly lubricated to give 100% service.

Clean and properly made non-corrosive, non-abrasive lubricants of proper characteristics for the application should be used.

Satisfactory lubricants can be had that are free from objectionable features.

A method of determining comparative abrasive properties and film strength has been illustrated, and we feel such a test should be standardized on for lubricants.

Lubricants that show up poorly on analysis will invariably not give best results in service.

DISCUSSION

PRESENTED AT CHICAGO SECTION MEETING OF A. I. & S. E. E.

DISCUSSED BY:

A. J. Whitcomb, Asst. Elec. Engr., Freyn Engineering Company, Chicago, Ill.

O. L. Maag, Lubricating Engineer, Timken Roller Bearing Company, Canton, Ohio.

H. R. Mathias, Engineer, Standard Oil Company of Indiana, Chicago, Ill.

J. B. Castino, Engineer, SKF Industries, Inc., Chicago, Ill.

E. W. H. Rennick, Lubrication Engineer, S. F. Bowser & Company, Fort Wayne, Indiana.

W. H. Oldacre, Director of Research and Engineering, D. A. Stuart & Co., Chicago, Ill.

G. T. Hollett, Combustion Engineer, Illinois Steel Company, South Chicago, Ill.

B. S. Okner, Chief Engineer, Ahlberg Bearing Company, Chicago, Ill.

T. E. Watson, Lubrication Engineer, Culbertson & Watson, Inc., Chicago, Ill.

A. J. Whitcomb: I was very much interested in Mr. Maag's paper and I want to express my own appreciation for the excellent talk he has given us. I was particularly interested in the tabulation of results, showing the results obtained with lead soap and other compounds. Mr. Marshall, our Chairman, is unfortunately not with us tonight but he gave me several questions that he asked to have brought up, hoping that some of the gentlemen present might discuss the matter and give us some information. The first question: What is the proper height to fill bearings with oil or grease? I noticed that one of Mr. Maag's illustrations shows the level should be slightly above the horizontal axis of the bottom roller.

Mr. Marshall brings up the point that a good many bearings are equipped with housings in which he can get only about a tablespoonful of oil or grease if they are filled to a level slightly above the axis of the bottom roller. Under these conditions it is hard to keep the oil or grease at the proper level and soon a bearing burn-out occurs there. If the oil level is to be so low, there ought to be a sufficiently large storage space provided.

Another point Mr. Marshall mentioned has to do with direct current motors in which the motor and possibly a gear unit are built together in a small frame. This brings the armature in close proximity to the bearing, particularly the bearing on the commutator end. The commutator may have a 50 deg. temperature rise. This heat travels to the shaft and then expands the inner race of the ball or roller bearing, resulting in burning out the bearing. Mr. Marshall has had to add additional clearance by turning down the shaft to get away from this trouble. Additional clearances have been found necessary to reduce the heating caused by expansion and binding; such as looser fit of inner race on shaft retaining, however, a tight fit, and also a greater clearance between rollers and races. It is believed the commutator and bearing on the D.C. motors should be placed farther away from the commutator to escape the difficulty.

Bearing builders should provide means for removing the bearing from the shaft. Ordinarily there is only a small space behind the bearing and this space is too small to permit prying off the bearing. Mr. Marshall has had trouble on 80,000 c.f.m. fans in which the oil storage space was so small that the oil was gone before he knew it, resulting in the burning out of the bearing and welding the inner race to the shaft. The bearing manufacturer was willing to give him a new bearing but he can't get the bearing off without possible injury to the shaft.

What are the means of measuring the height of the lubricant in the bearing while the bearing is operating? In bearings that have small storage space, the agitation is so great, one cannot tell much about the lubricant level. If the storage space was larger, a more accurate level determination could be made. What are the bearing people doing in this matter?

O. L. Maag: That is largely an engineering problem. There is always an ideal to reach, and I know our engineers are always trying to come nearest to that ideal, but at times they can only approach it, due to limited space and operating conditions. An example of taking care of the oil levels with sufficient lubricant was illustrated with railroad bearings. In motors, however, the oil level will be lowered when the motor is started and allowance must be made for it. This problem we feel comes under design, and it is really up to the designing engineer to take care of the applications. Our engineers have these thoughts in mind and are always trying to take care of lubricating conditions as they meet them.

H. R. Mathias: Some of the remarks I have prepared will probably appeal more to some of the oil men here than to some of the other members, inasmuch as they refer particularly to technical matters relating to lubricants. Some of them will be questions we wish to have Mr. Maag clear up while he is here. I think the things which are of interest to the oil people with respect to E.P. lubricants, however, are worthy of pointing out at this time.

Before making any general remarks, I think every oil man here will agree with the statement made relative to keeping in mind abrasiveness and stability as well as film strength in E.P. lubricants.

I wish to suggest to Mr. Maag that the load carrying capacity of some greases is lower than that of the oils from which they are made because of the type of soap contained, rather than because of inferior cooling from the grease during the test run. Based on the work which we have done, the film carrying abilities of various oils may be consider-

ably varied by the types of soaps used in the greases. For instance, calcium and sodium soaps in greases seem to reduce the load carrying properties of the E.P. oils contained, apparently by chemical action on the extreme pressure film, whatever it may be. Other soaps of some types do not have this action.

The apparent advantage in load carrying properties of some lubricants made from paraffine base crudes as compared with Mid-continent and napthenic crudes, we feel is primarily due to inherent properties of the lubricant with respect to the elusive property known as "oiliness", rather than to differences in viscosity-temperature relationships. In this connection, I have in mind results with a synthetic oil with which we have been working which are much inferior in load carrying ability to lubricants from Mid-continent sources, although the synthetic oil is much superior in viscosity-temperature relationship to paraffine base oils. I might say, however, that the service results on such an oil show it to be considerably superior to other types of lubricant.

I think we should pause here a moment and reflect upon the application of extreme pressure lubricants in steel mill and other service. Doubtless, owing to the interest created in E.P. lubricants by such discussions as these, a great many extreme pressure lubricants will be applied in places where they have no business. It is a well-known fact that these extreme pressure lubricants, owing to their properties, are more abrasive than the straight mineral products. There are many places around the mill, around an automobile, and other places where an extreme pressure lubricant is not necessary. I would caution you not to apply extreme pressure lubricants to points where they are not needed. I think Mr. Maag will agree with me in that respect. A great deal of damage can be done by so doing.

The development of extreme pressure lubricants is still in its infancy. I have some hesitancy in making any comments on Mr. Maag's paper as I do not feel that we oil men are in a position at this time to say a great deal about it. We have just scratched the surface but I think we will eventually work out the difficulties that now appear.

I should like to ask Mr. Maag a question relative to the operation of the machine. We have found some very interesting facts with regard to it. Considerable variations in the degree of wear on several tests with the same lubricant from the same batch have resulted. I am unable to ascertain whether this was accidental in the few tests we observed but wish to inquire in that connection relative to the uniformity of the test blocks which are usually supplied for this work. For example, with one lubricant in one case under a 33 pound load, the width of the line was 0.040 in. and in another case with the same lubricant the width was 0.030 in. This does not seem to be a great difference but the unit pressure in one case is figured to be 17,700 lbs. and in the second case, 23,500 lbs., a difference of 5,800 lbs per sq. in. in the actual load. We have run across several occurrences like this, enough to make us want to know a little about it.

I should like to get Mr. Maag's idea of the cause of this variation, and also to get his opinion of the actual mechanism of extreme pressure lubrication. What goes on at the boundary between cup and disc or the roller and race? There seem to be as many ideas as there are people consulted on this.

I have just one more question. In the table showing the wear results with various lubricants, Lubricant No. 6 plus 10% graphite shows a satisfactory load of 85 pounds, whereas the same lubricant plus 10% zinc oxide shows a satisfactory load of only 14 pounds. Was that a typographical error?

O. L. Maag: Yes, it is a typographical error and should be 85 pounds and not 14 pounds as shown.

I will try to give you an answer; however, your ideas as to just what goes on at the boundary with extreme pressure lubricants are as good as mine, but I really do not know. The common opinion seems to be that there is a chemical reaction with the lead soap sulfur oils forming a film on the bearing surfaces which carries the loads. However, in the case of No. 6 Lubricant (a lead soap sulfur chlorine compound) it might be that a solvent is used which carries the lubricant film under the bearing surfaces, as there is apparently no more wear than with a normal non E. P. Lubricant. The results illustrated with dichlorethylether might be used to back up this idea. I, however, do not know what actually happens.

The variations in wear tests mentioned, Mr. Mathias, are quite interesting. We have not done enough work to be able to say what variations in results are permissible. Some variations may be attributed to variations in test blocks and cups; however, each piece is tested for hardness before it is sent out, so that this variable is kept at a minimum. We also endeavor to keep the ground surfaces as near a uniform smoothness as possible, as we know it affects the results. In regard to the width of the scars on the test blocks spoken of, figuring 19,000 pound load per square inch in one case and in another, 23,500 pounds, I will present you data given me last week by a representative of a large oil concern. They ran several oils for abrasion for a period of time, and each oil abraded until the final unit loading came to a constant, for one oil it was 23,500 pounds per square inch, for another, 17,000 pounds per square inch, and for another, a different definite unit loading. We do know that with abrasive oils the most rapid wear is at the start and we feel that a test should be run over a period of time to allow for variations in test procedure and so that better check results may be obtained.

There is lots of work to be done along this line, and we feel we have just started; perhaps, in six months or a year, we will really have some definite idea.

J. B. Castino: Answering Mr. Whitcomb's question relative to the oil reservoir in a bearing housing, we find in designing housings for anti-friction bearings that it is necessary to design the housing with sufficient oil capacity and also to remove the lips of the housing far enough from the seat of action between the bearing and the oil. If the lips are close to the seat of action, the bearing will pump out the oil through these apertures. If you have a housing that is of small capacity and the lips of the housing are close to the bearing and the speed is relatively high, the oil will gasify and disappear through the housing lips. In this type of housing the lips should be removed as far as possible from the seat of action and sufficient oil capaci-

ty designed into that housing, incorporating proper flinger design protection.

Relative to the extraction of a bearing from the shaft:

If it is difficult to obtain proper purchase on the inner race of a bearing, we incorporate in designs, what we term as "pull-off collars" so you can get sufficient purchase on the inner race of the bearing to extract it from the shaft. Mr. Whitcomb also spoke of the temperature rise in the commutator and the commutator bearing and stated that he corrected this by making a loose fit between the bearing and the shaft to take care of the temperature rise. We do not believe that this is advisable. If you have a temperature rise in a shaft and it is transmitted to the bearing, you should incorporate a bearing that has enough internal looseness to take care of that condition. Otherwise, you will have a peening action between the race and the shaft and ultimate destruction of the bearing and the shaft. If the temperature rise is abnormal and cannot be taken care of in this manner, we recommend cooling the shaft by means of water cooled end caps.

Mr. Whitcomb also spoke of a bearing application in a fan. When a bearing is in the draft area of a fan and it is not sufficiently protected with flingers to divert the suction of the air, there is a possibility of siphoning the oil from the housing. A special designed housing should be used to defeat this action.

How do you keep a constant level in a housing by your device when the tendency of the bearing, when rotating, is to draw the oil level at an angle in a direction of rotation?

Visualize a horizontal oil level in a housing. A horizontal shaft. When a bearing begins to rotate, the oil draws up at an angle in the direction of rotation and depending upon the speed and quantity of oil in the housing. I would like to know how you can indicate an oil level when conditions like this exist.

E. W. H. Rennick: Mr. Maag's paper contains some very interesting facts especially relative to the loads the various oils will carry. The development of this new test represents another step nearer to that which so many investigators are seeking. The finding of a positive method of comparing one oil with another which will give some idea of how each will be expected to perform in a roller bearing. It may lead to the development of lubricant formulae hoping to meet the various load conditions, thus doing away with so many personal opinions and guess work.

In reviewing the field application we find high duty roller bearings in steel mills lubricated with anything from grease to ordinary bearing oil. The choice of lubricant is controlled somewhat by the possible loss of oil or the high contamination of the lubricant with water. It is therefore necessary to provide well made means for retaining the oil within the bearing and at the same time preventing the water from entering.

The design of the housing around the bearing very often determines the application of a liquid lubricant. If there is not sufficient space between the bearing and housing the oil will be picked up by the rotating rolls and cage and not drain from the bearing causing the oil to leak out. It has been found where a bearing has ample space for the oil to drop to the bottom of the housing, after it has passed between and lubricated the rolls, a copious supply of clean cool oil can be fed to the bearing thus washing out all gritty particles and materially increasing the life of the bearing.

The use of well protected vents at the tops of the bearing housings will also help the free drainage. The size of the drains should be ample taking into consideration the change in viscosity due to changes in temperature.

The use of flingers is to be determined by the speed. Flingers will not function at low speeds, it is therefore necessary to use a stuffing box or wiper arrangement to retain the oil.

W. H. Oldacre: I don't know that we have anything further to add. I would like to ask Mr. Maag whether the weights given are the total weights of the cup and the block. We have found that there is a tendency for metal removed from the block to adhere to the cup, and I am wondering if that has been taken into consideration.

Another thing that I think we should call attention to is that the load varies with the width of the groove. This is a question that we have always had with us, but if we stop to analyze the situation we find that we have a cylindrical and a plane surface in contact, and therefore if my mechanics is accurate we should have an infinitely great load for the first few seconds of operation, at least. With that in mind, it is apparent that the lubricating film, or whatever it is that we are depending upon for lubrication, must be ruptured for the first few seconds or until we get a bearing surface.

In our experience, when testing petroleum lubricants which break down under a low load, a great deal of metal is removed in a few seconds. It would seem that it is impossible to control this closely enough so that we can get data regarding the width of the bearing during the running in period.

G. T. Hollett: I was very much interested in the paper and also the comments. We have had considerable experience with the Timken Lubricant Testing Machine, and find that the variations in load carrying capacity with various greases is a function of the grease temperature. We find that if tests are conducted for several hours that we can secure results that can be duplicated. If short tests of a few minutes are conducted, it is difficult to get a series of tests to all show the same result.

With extreme pressure greases tested near their maximum carrying capacity, we find a very rapid increase in temperature until the area of bearing has worn until the unit load per square inch has dropped very materially. During this period of high temperature the bearing is wearing rapidly. After the area has increased sufficient the wearing stops and the temperature drops to normal bearing temperature.

B. S. Okner: In reply to the questions relative to compensation for expansion of shaft and inside ring of an anti-friction bearing to avoid overloading and excessive temperature rise in operation which might cause an anti-friction bearing to burn up, would say that this can be easily provided for by installing ball bearings with an internally loose fit. This is accomplished by allowing lateral play

of from eight to twelve thousandths of an inch (.008" to .012"). A bearing so assembled should be mounted with a light press fit on the shaft, and a sucking fit in the housing.

As to roller bearings, care must be exercised to allow only a limited amount of looseness as excessive looseness has a tendency to cause the rollers to chip, and affect the life of the bearing; however, due to the present skill of mechanics, and information supplied on correct mounting of bearings, there is only slight possibility of misadjustment or improper fitting.

With regard to the amount of lubricant to be provided, and the means of retaining it in an anti-friction bearing housing, the general practice is to fill the housing with oil to the level of the center of the bottom ball or roller. The housing should be designed with sufficient capacity so that at the level specified an adequate volume of oil is provided. Thus enough lubricant will be supplied for a considerable period of time, and heat, due to internal friction or churning of lubricant, which is in many cases the cause of high operating temperatures in anti-friction bearings, will be prevented. Where grease is used as a lubricant, fill housing one-third (1/3) full.

There are many methods to retain lubricant under varying shaft speeds. In addition to the various plain and labyrinth type housing closures, we have the standard lubri-seals employing leather, cork, asbestos, and felt.

At moderate operating speed, lubricating once in twelve months has been successfully employed without detriment to the bearings, although recommended practice is to add lubricant every three to six months, and at the end of twelve months to flush housing with kerosene and refill with fresh oil.

With regard to the third question relative to provision for, and method of removal of bearings from shaft in cases of burning, the manufacturers of ball and roller bearings will be more than glad to assist in designing anti-friction bearing applications and submit specifications as to the size of shaft shoulder needed relative to the outside diameter of inner race so as to provide sufficient clearance to permit the bearings to be removed from the shaft; however, each machine designer can consult the S.A.E. Handbook or any ball or roller bearing engineering data book for these dimensions.

I do not share the same opinion as to the advisability of one lubricant adaptable to lubricating both the gear set and anti-friction bearings for the simple reason that this will encourage designers to include gears and bearings in one housing, which is not accepted practice in anti-friction bearing applications despite the fact that there are many units thus constructed.

Bearing housing should be sealed from gear housing, and lubricated separately, as fine chips, created from gear friction, combining with the lubricant, form a grinding compound which, due to constant agitation, finds its way into the bearing housing and causes excessive wear of the bearing, and in some cases a bearing lock. I believe that the additional cost of sealing bearings from gear housing lubricants and the cost of maintenance in lubricating bearings once in three or six months will be more than offset by the savings due to increased life of the bearing.

A Member: As we have some of the bearing manufacturers and oil people here, a question came to my mind and that is,—it seems there is some question as to the advisability of using the circulating system with filters, at least with certain grades of oil, as you are filtering out some of the residue and changing the composition of the oil. I wonder if it would be advisable to either eliminate the filters in the circulating system, or would it be advisable to use a straight oil that had no foreign compounds such as sulfur in it. Also, is there a possibility of abating this breakdown by cooling the oil to hold the temperature down?

O. L. Maag: I feel your point is well taken regarding cooling the oil. The breakdown of an oil is high with temperatures and pressures you often encounter. If you can keep the oil cool, it will not oxidize or break down nearly so readily. I will illustrate the advantage of cooling along another line. In a cutting oil unit, temperatures rose at the end of a day's run to around 130°F. to 140°F. and we found variations in gauge of material. By putting on a cooling system, and keeping the oil temperatures around 80°F., we obtained around 10% more production, due to better and cooler working conditions, and the oil does not break down nearly so much. Similar results will occur on bearing applications. If you keep your bearings and oil cool you do not allow the oil film to become thin enough to break, and it will not oxidize or break down rapidly when kept cool.

A Member: I don't believe the use of the filter will alter the composition of the oil to any extent. It simply removes the dirt and deposits rather than changing composition. That is already changed.

T. E. Watson: There has been quite a little mention of the lack of a lubricant reservoir in anti-friction bearings and several of the gentlemen have mentioned constant level lubricators. There are quite a number of installations which are working out very successfully. There is another question about the tendency of the rotation of the bearing to draw the lubricant and raise the level. This may also be overcome.

J. B. Castino: How do you keep a constant level in your device when you have oil in the housing and, to begin with, the bearing begins to rotate and throws the oil up?

T. E. Watson: There is not enough oil in the bearing to cause the rotation to draw up any appreciable amount of oil. The amount of oil that can be drawn up is of such a small amount that it does not affect it.

J. B. Castino: You can visualize a horizontal level of oil in a housing. When that bearing rotates the oil draws up and commences the rotation. I would like to know how you indicate that oil level after it begins to raise off the horizontal level in an angle.

DISCUSSION

PRESENTED AT PITTSBURGH SECTION MEETING OF A. I. & S. E. E.

DISCUSSED BY:

J. Farrington, Electrical Supt., Wheeling Steel Corp., Steubenville, Ohio.

O. L. Maag, Lubricating Engineer, Timken Roller Bearing Company, Canton, Ohio.

C. L. Johnson, Chief Chemist and Technical Supt., Franklin Oil & Gas Co., Bedford, Ohio.

J. L. Brown, Mechanical Engineer, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

H. H. Wood, Engineer, Timken Roller Bearing Company, Canton, Ohio.

J. L. Young, Engineer, Timken Roller Bearing Company, Pittsburgh, Pa.

M. Reswick, Chief Engineer, Pennsylvania Lubricating Company, Pittsburgh, Pa.

H. A. Smith, Engineer, Vaccuum Oil Company, Inc., Pittsburgh, Pa.

C. B. Karns, Engineer, Standard Oil Company of Pennsylvania, Pittsburgh, Pa.

W. E. Smith, Lubricating Engineer, The National Refining Company, Cleveland, Ohio.

R. R. Owen, Engineer, General Electric Company, Wheeling, W. Va.

J. Farrington: Relative to speed on anti-friction bearings, if we have a speed of, say 300 R. P. M. that is giving perfectly satisfactory results with an ordinary grade of oil, what increase of speed could you obtain without getting into trouble? There are several variables in there, but could you give some idea of what the limits would be.

O. L. Maag: It is hard to say. The permissible increase will depend on how close the product of your load and speed is to the film strength of the oil used. We do not have enough data yet to tell the story accurately. The curve you saw has, however, given us information as to about when to look for trouble in heavily loaded applications when speeds are to be increased.

J. Farrington: Was that a predetermined curve?

O. L. Maag: No, it was made from data obtained on the Lubricant Tester on cylinder oil and a lead soap compound.

J. Farrington: Is that a fixed ratio between the speed and load?

O. L. Maag: The curve indicates that.

J. Farrington: You really can go up on the speed without getting into trouble.

O. L. Maag: Yes, to a limited extent with light loads.

C. L. Johnson: Have you found any difference in the character of the sulfur?

O. L. Maag: Yes, sulfur that is found natural in the oil in our experience helps to increase film strength. A certain Smackover with 3% sulfur and a viscosity of 150 seconds at 210°F. gives a ratio of 2 to 1 over a cylinder oil of the same body, however, the sulfur may not have been all combined. If you add about 2% free sulfur you will obtain approximately a 3 to 1 ratio. If sulfur is combined with fatty oils you may obtain a 4 or 5 to 1 ratio or higher. Another angle you probably noted is the abrasiveness of sulfur. By adding sulfurized lard to mineral oils the abrasiveness of the product will fall some place between free sulfur and natural sulfur found in the oil, indicating that the form in which sulfur is present governs how abrasive the lubricant will be.

A Member: Would the abrasive action of these various oils affect the Timken bearing more severely on the end of the taper roll?

O. L. Maag: The ones that show abrasiveness in the Tester produced wear on the rollers and the roll ends. Free sulfur, for example is a fairly bad one. Another highly abrasive material is asbestos fibre. There are quite a few lubricants on the market that contain asbestos fibre because it reduces gear noises. We had to discontinue one test of a lubricant containing asbestos on account of the excessive wear, which proved to our satisfaction that asbestos is very abrasive.

J. Farrington: Have there been any installations of anti-friction bearings in crane service that have oil cells? I inferred that your characteristic picture represented a bearing in the housing of stationary apparatus, and wondered if that had been applied to any cranes.

O. L. Maag: I believe there are a few applications where they use oil. I understand that with constant level oil bottle, oil works very well. In these installations, the tendency is for the bearing to throw the oil to the outside, so you must make allowance for that when regulating the oil level. In installations such as ring oilers in electric motors the constant level oil bottles work very well and I am confident would prove very well on crane bearings.

A Member: What kind of oil was used in the Timken Roller Bearing Locomotive—was it oil or grease?

O. L. Maag: Steam refined cylinder oil, 125 to 150 sec. viscosity at 210°F. It is working very satisfactorily on this locomotive. The above holds true also for Pullman and freight car roller bearings. We desire the heavier oils as we have less oil loss, hence the oil costs are thus kept to a minimum. For railroad applications we would like to have an E. P. Oil (extreme pressure, compounded oil) of zero cold test or below, with a viscosity of around 100 seconds at 210°F. and a load capacity of 15 pounds for an all year-round lubricant. Such an oil would be satisfactory in the Northwest where they have temperatures of 20 to 30 degrees below zero; also in the South where temperatures go as high as 110°F. We hope that in co-operation with the railroads and oil companies year-round oils for those applications will soon be available. We now have three such products ready for marketing by three major oil companies and expect several more to be available in the near future.

J. L. Brown: I was very much interested in listening to Mr. Maag's paper. It is particularly interesting to me to see that the company which is making a device to supplant film lubrication is now becoming the pioneer in advancing this art. The papers a few months ago came out with a news item that emanated I believe from Penn State College and had a title something like this: "Who was Atlas anyway? See what Oil can support." Then it went on to show how many thousand pounds a small area could carry. Of course being interested in motor design particularly, I am visited by various people making anti-friction bearings and they would like me to supplant the oil film bearing by a bearing using rollers or balls, and attempt to throw out this oil film.

Before we had roller bearings at all we managed after a fashion with what now appears to be inferior lubricants from the standpoint of film strength but now it seems after we have gotten roller bearings we have to have an oil that is from three to six times better than we ever had before we had roller bearings. So I should like to inquire, if Mr. Maag will be so kind as to tell us, just wherein we have failed with the introduction of roller bearings in supplanting the oil film; and where therefore we need this very good oil? We have been getting along after a fashion with these oils that have been given us in the past. It seems there must be some trouble of some kind, perhaps some trouble with the bearings, that makes it necessary to put in these very good oils. I should like to have a distinction made as to where we could perhaps get along with the older types, and just exactly where we ought to have the better type.

O. L. Maag: You are of course aware that speeds and loads are much above what they were in the past. For the ordinary speeds that you have had in the past and on small bearings, we have no trouble with the ordinary oils. It is only for increased speeds and loads that we have had to increase the E. P. quality of the oil. In the gears of automobiles we had unit loads maybe one-half of what they are today and speeds were below present speeds, and on these ordinary cylinder oils were entirely satisfactory. It is a case of increased loads and speeds that has brought this about. I know of no cases where they have not increased loads or speeds where an extreme pressure oil is required.

J. L. Brown: That was not quite as specific an answer as I was looking for. Are there any cases of sleeve bearings where we require this particularly good oil? To what extent have you gentlemen succeeded in doing away with the oil film by putting in rollers? There are perhaps certain parts that would not need the oil. On that conical surface, perhaps we wouldn't need it, but when it comes to the contact of the ends of the rollers with the adjacent rib on the inner race, perhaps we would. You, of course, are in the taper roller business, but you know a good deal about ball bearings and other types of roller bearings. It would be of practical value to know what parts of any of these bearings need this particularly good oil, and if there are some bearings that do not have need of the particularly good oil, so that when using them we could get along with the older types of oils.

O. L. Maag: Where you haven't increased your loads and speeds you could say you don't need E. P. lubricants on anti-friction bearings, or other types.

J. L. Brown: When the balls or rollers roll free, do you need this particularly good oil for such rolling contacts; or is it required only for the sliding contacts?

O. L. Maag: I could not say just what occurs that demands the lubricant. We know you need lubrication, whether you have plain bearings or ball or rollers. There must be some slippage in all cases that require lubrication. You can classify all bearings alike in the respect that they all must be lubricated. The only places we have encountered where we need an E. P. oil on anti-friction bearings are where we have exceptionally high loads and high speeds.

J. L. Brown: We had a plain bearing on an elevator motor where we were getting much heavier loads than previously, and the bearing pressure was something like 450 pounds per square inch. It did not seem to make much difference on the oils we had used before until we reached the speed at which the elevator runs for a moment or two before the brake sets. At this point we did find that one of the oils we had been using gave us rather too large losses, indicating the film was breaking down. We had made some tests on different oils and discovered in the case of a certain heavy automobile engine oil and castor oil that the coefficient of friction did not go up rapidly at the low journal speed of 7 feet per minute as in the case of the others, but continued at about the same figure. We then used this oil (castor oil being less convenient to obtain) and we found it did make quite a saving. That was not a case of high speed, but a heavy load. Would this improved grade of oil—this compounded oil—act as this oil did on even greater loads at very low speeds?

O. L. Maag: I think sulfur oils would help you. Thinner oils would cut down friction losses.

H. H. Wood: I can perhaps add some information on a definite job which will aid in answering Mr. Brown's questions. There was recently built a very large roughing mill drive whose motor pinion shaft was 20" diameter and from 25 to 30 ft. long. On this shaft were mounted two 50 ton flywheels. The operating speed of this shaft is about 380 R.P.M.

On account of the high speed and the heavy bearing loads which would be encountered, the mill builder felt that it would be impossible to carry them on plain oil lubricated bearings and it was decided that anti-friction bearings were the only type which would get away with this job. Bearings 20" bore by 33" O. D. built by our company were used. On account of the heavy loads encountered and extremely high speed of these large diameter bearings, we would say that this is a place where an extreme pressure lubricant should be used.

This was done and much thought was given to the design of the lubrication system. A satisfactory operating temperature was obtained by a circulating oil system which not only pumped the oil to the bearings but also sucked it away which eliminated heat which would result from churning of an excessive amount of oil.

In designing bearings for this job not only our company but other bidders gave careful consideration to the high peripheral speed of the rollers in the

bearings, designing them in such a manner that the R.P.M. of the rollers themselves was kept to a minimum.

J. L. Young: As I understand it, Mr. Brown, your question is: "Do you need this special lubricant in electric motor bearings?" I believe Mr. Maag will agree with me in saying the answer is "No".

J. L. Brown: Sometime ago one of you gentlemen sent around word that we dare not use some of your bearings that go in the auxiliary mill motors above a certain speed and particularly ruled out grease, as I remember, the trouble being overheating. We got something like 50° C rise and that was considered unsatisfactory. I think that at the last meeting of the Special Bearings Committee of the Association of Iron and Steel Electrical Engineers, it was decided to send around a questionnaire about these bearings as to how fast they could run; how fast with oil, and how fast with grease. Because of our experience with lubrication at the speed at which we sometimes run the larger bearings, the question arises in my mind: would we be any better off with this compounded oil in those particular bearings?

O. L. Maag: Did you have galling? Extreme pressure lubricants are compounded to prevent galling or cutting of bearing metal surfaces. If it is a case of heating from oil churning, lower the oil level, or better still, install a circulating oil system. E. P. oils will prevent galling in installations where, with the ordinary oil, due to the speeds and loads encountered, galling or scoring occurs.

J. L. Brown: It is then for pressure combined with speed, and not for speed alone?

C. L. Johnson: A paper presented by Doctor M. Dover before the Petroleum Division of the American Chemical Society at Tulsa in 1926 is of interest in this connection. She describes the elaborate precautions taken to remove the oil films from the test surfaces of the Deeley machine and in a discussion indicated that compounded oils containing fats such as olive and lard oils were so difficult to remove, even with organic solvents, that it was necessary to regrind and polish the surfaces between tests.

This increased oiliness or absorption to the metal surfaces and decreased coefficient of friction of the compounded oils indicated that they would carry increased loads.

Before this same meeting, W. G. Wilharn read a paper in which he reported a number of slider determinations using a brass plate and steel slider lubricated with several typical lubricants. The analysis of his results show that the lubricating value of the oils tested is not shown by the usual tests of viscosity, flash, fire, etc., but depends on the elusive property of oiliness or absorption to the metal.

In partial answer to Mr. Brown's question, I would cite Wilharn's paper to the effect that electrical machinery bearing pressures have become so high that he doubted if a full film is formed and in such cases, the property of oiliness of the lubricant is the all important factor.

Such properties as extreme oiliness cannot be obtained from straight mineral oils but from "doped" lubricants as shown in Mr. Maag's work.

Maurice Reswick: Mr. Maag has given us in his paper a very able presentation of the latest developments in lubricating oils and greases, and his story reads very much the same as that of the development of alloy steels. First we had wrought iron and carbon steel, but gradually other elements were added until almost any desired characteristic may be imparted to steel of the proper alloy. Similarly, in the field of lubricating oils and compounds, starting out with refined mineral oil and by incorporating in it certain ingredients like lead oleate, chlorine, sulfur, etc., it appears that we can "build up" a lubricant that will measure up with previously determined specifications and desired characteristics, especially as regards film strength and reduced wear on the surfaces to be lubricated.

Our theories on lubrication are also being gradually revised in the light of the latest observed facts in actual performance. Not so long ago we have believed that the function of a lubricant is to keep two rubbing surfaces apart and to prevent metal-to-metal contact. This purely mechanical conception does not hold good in the case of boundary lubrication, that is, where the oil film is subjected to pressures so high that it is squeezed out or thinned down to almost the vanishing point. We are beginning to recognize now that certain interactions of a chemical nature might take place between the lubricant and the metallic surfaces of the journal and bearing, or rollers and races in anti-friction bearings, so that the interposition of a fluid oily film is not altogether essential to prevent seizure. The adsorption of the lubricant on the surface of the journal or other bearing elements results in the formation of such compounds like iron-oleate or iron-sulphide, a very minute, submicroscopical coating of which serves the purpose of preventing pure metal-to-metal contact.

The complexity of what actually takes place at the rubbing or rolling surfaces when a lubricant of the extreme pressure type must be used in boundary lubrication, indicates that a certain amount of caution must be exercised in the choice of these newer lubricants. By the addition of certain ingredients and the proper choice of materials it is possible to produce a lubricant that will withstand any desired operating pressure up to the point where the metal itself will begin to flow or fail under compression, but this advantage is often gained at the expense of some other equally desirable characteristics, such as freedom from corrosion and stability.

The ideal extreme pressure lubricant represents a compromise of high film strength with the least amount of abrasion, corrosion, tendency to separation and emulsification. It is well to recognize also that while laboratory tests give advance indications of the properties of lubricants and are extremely useful for comparative purposes, the real test in the final analysis is satisfactory performance over a long period of time in actual operation.

H. A. Smith: In connection with E. P. lubricants for gears, it is generally accepted that these chemicals such as sulfur or chlorine, either singly or combined, form many compounds which prevent scoring of metal surfaces when rubbed at high pressure. The action seems to be chemical where the metal is attacked below the scoring temperature; the microfilm acting as an anti-welding flux to prevent a seizure. In other words, "scuffing" or "gall-

ing" which occur when the surfaces break thru the mineral oil film, producing sufficient localized heat to make a forge, is eliminated.

In the manufacture of these materials, the objective is to devise compounds which are stable at ordinary temperatures and decompose into active bodies when and if needed. None of the materials now used completely fulfill these conditions and to the extent that these compounds decompose unnecessarily, they become dangerous to the machine as a whole.

C. B. Karns: I am pleased to note that my former associate—Dr. Maag—has proven to his satisfaction that lead oleate when properly manufactured free from lead oxide and foreign material will carry the necessary load without being abrasive or showing any wear.

A Member: Do you have to use a large quantity of that lead oleate in order to get high film strength? Is it easily mixed or united with the oil?

O. L. Maag: It is a matter of compounding and a matter of quality, or properties of stock used. Have a sample that came in this week that had 14% of lead soap as lead napthanate, ½% sulfur, no chlorine. It carried a 60-pound load which is exceptionally high. That may refute the idea many of us had that it is the sulfur in lead soap products that carries the load, however, not a lot of sulfur if properly combined is necessary. Of course, many lead soap compounds have high sulfur. We have added lead oleate to a straight oil, and while we obtained some increase in film strength, did not secure what we should have for a good E. P. lubricant. I feel that the quality and nature of stock enters into it to a large extent so that the oil companies still have a lot to do before the question is solved.

W. E. Smith: "What was the viscosity of the stock oil and the percentage of lead oleate used to secure the best tests as shown in the table giving the result of tests for load and abrasion?"

O. L. Maag: The amount of lead oleate is likely governed by the stocks used and method of compounding, usually 5% to 10% is used. The test referred to contained 5% lead oleate with a refined cylinder oil. We did not run this for load, the test being conducted to prove that lead oleate itself was not abrasive. Other tests indicate that the refinement of stocks has something to do with the effectiveness of sulfur and chlorine. We tried adding a sulfurized fatty oil also a chlorinated product to a black oil and did not obtain very high load capacity oils; then took a filtered stock with the fatty oil and chlorinated product and obtained twice the load carrying capacity. With certain unrefined stocks deterioration of the load capacity occurs quite rapidly, while if refined stock oils are used, the load capacity of the oil holds up over a much longer period of service.

R. R. Owen: What are the principal chemical and physical properties of lead oleate?

O. L. Maag: Soaps, in brief, are combinations of metals, usually lime, soda, or lead with the acid radicals of fatty oils. If we treat lard oil with ordinary lime we obtain a calcium soap; with lead oxide we secure a lead soap, etc. Lead oleate such as we used was about the constituency of ordinary toilet soap and had a brownish color.

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ITEMS OF INTEREST

PERSONNEL CHANGES

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Chas. F. Stone, formerly vice president of the Atlantic Steel Co., has been made president of the Atlantic Steel Co., Atlanta, Ga., succeeding Robert Gregg who has been appointed vice president in charge of sales of the Tennessee Coal, Iron & Railroad Co., Birmingham, effective Aug. 1, succeeding Willard Wilson, who has resigned as vice president and general sales manager.

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Edward R. Williams has been elected president and a director of the Vulcan Mold & Iron Co., Latrobe, Pa., succeeding his father, the late Col. Edward H. Williams. The younger Mr. Williams has served as vice president of the company for several years, and his position has not been filled. H. H. Smith continues as secretary and treasurer of the Vulcan organization.

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James A. Farrell, former president of the United States Steel Corpn., has been elected a member of the board of directors of the Chamber of Commerce of the United States.

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George F. Benson, former president, Monongahela National Bank, Pittsburgh, formerly treasurer, National Tube Co. and vice president of the Union Trust Co., Pittsburgh, has been named treasurer of the Youngstown Sheet & Tube Co., Youngstown, O. He succeeds Walter E. Meub, who was secretary-treasurer until recently elected vice president in charge of the company's finances.

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Walter E. Meub, whose election as vice president of the Youngstown Sheet & Tube Co., in charge of finances was recently announced succeeded William J. Morris, who had been made president of the Continental Supply Co., pipe distributing subsidiary of the Youngstown company, with headquarters at St. Louis. J. C. Argetsinger, general counsel of the company for two years, has been elected secretary.

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E. C. Smith has been appointed chief metallurgist of the Republic Steel Corpn., Youngstown. He has been for some time assistant district manager for the Central Alloy district of the corporation at Massillon. His headquarters hereafter will be located at the main offices in Youngstown.

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The Ideal Commutator Dresser Co. has appointed

Mr. R. C. McDaniel as their representative in the State of West Virginia, in charge of sales of both their Motor Maintenance Equipment and Wiring Device lines, handling Commutator Resurfacers, Grinders, Mica Undercutters, Blowers, Thread Lugs, etc., Wire Connectors and tools. Mr. McDaniel makes his headquarters in Parkersburg, W. Va.

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Mr. S. S. Wales, Chief Electrical Engineer, Carnegie Steel Company, Pittsburgh, Pa., has retired from active duty, August 1, 1932.

Mr. Wales graduated from Rose Polytechnic Institute in 1891 and started to work with the General Electric Company in 1892 at Lynn, Mass. In 1893 he entered the employ of the Carnegie Steel Company under Mr. A. C. Dinkey and in 1894 accepted a position with the Ohio Steel Company as Electrical Superintendent. After five years service with the Ohio Steel Company Mr. Wales was recalled in 1899 and made Electrical Superintendent at the Homestead Works. In 1905 he was made Superintendent of the Armor Plate Department, a position which he retained until 1915.

Mr. Wales spent the greater portion of two years in Europe in special metallurgical work.

In 1916 Mr. Wales was appointed Chief Electrical Engineer of all the Carnegie plants and held this position until July 31, 1932.

Mr. Wales is an active member of the Association of Iron & Steel Electrical Engineers and was elected President of this Society in 1927. He served on many of the National Committees, contributed many valuable papers and was chairman of the committee that prepared the Specifications for Heavy Duty Electric Overhead Traveling Cranes, which are used throughout the Iron and Steel Industry.

Mr. Wales will make his future headquarters in the City of Pittsburgh.

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Otto H. Falk, President of the Allis-Chalmers Manufacturing Company, since 1913, has been elected Chairman of the Board of Directors of that Company and Max W. Babb, Vice President, has been elected President to succeed him.

At the same time, W. A. Thompson and Wm. Watson were elected Vice Presidents; R. Dill, Secreary and Treasurer; J. A. Keogh, Comptroller; and H. W. Story, General Attorney. All of these officers have been connected with the Company for many years.

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The National Carbon Company, Inc., have transferred to their Pittsburgh office Mr. M. B. Farrar to handle their carbon brushes and Gredag lubricants in this territory. Mr. Farrar was formerly the National Carbon Company's representative at Birmingham, Ala.

Mr. J. B. Hayes has been put in charge of the Birmingham office of this company, taking Mr. Farrar's place, located at 1010 S. 22nd Street, Birmingham, Ala.

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PROSPERITY PARAGRAPHS

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The United States Government, Department of The Interior, Bureau of Reclamation, has just awarded a $10,908,000 contract to the Babcock & Wilcox Company, 85 Liberty Street, New York, for the fabrication and installation of the welded plate-steel outlet pipes for the hydraulic power and flood control tunnels at Hoover Dam. The contract involves such unusual features as the fabrication of the largest diameter pipes ever constructed, from plates about three inches thick, and the erection of a plant, built solely for the purpose, on the Federal Reservation near the Dam. The total length of the pipes to be constructed exceeds twelve thousand feet with variations in diameter from 8½ to 30 feet.

This is the second largest contract of its kind ever awarded by the Department of The Interior, and makes available to the steel industry about 55,000 tons of plate steel, the largest tonnage actively considered so far this year.

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The contract awarded to Combustion Engineering Corporation in March of this year by the United States Treasury Department, Washington, D. C., covering 4 complete steam generating units for the new Triangle Heating Plant, contained an option to purchase two additional duplicate units. The government has now exercised this option, and Combustion Engineering Corporation has received the contract for the two additional units.

These units, like the four on the original contract, are designed for 400 lb. pressure and are guaranteed for a continuous production of 215,000 lb. of steam per hr. and a maximum production of 237,000 lb. per hr. from and at 212 deg. fahr.

The contract includes boilers, stokers, steel-encased boiler settings, water-cooled furnaces, and forced and induced draft fans and drives. The stokers will be of the multiple retort, underfeed type, 15 retorts wide by 45 tuyeres long, and will have a projected grate area of 480 sq. ft. each.

United Engineers & Constructors, Inc., Philadelphia, are the consulting engineers.

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As part of an electrical project calling for the expenditure of 8½ million dollars in the next two years, a contract of approximately 1¾ million dollars has just been awarded to the Westinghouse Electric and Manufacturing Company by the Philadelphia Electric Company for a 183,000 kv-a. turbine generator to be installed at the Richmond Generating plant, on the Delaware River, Philadelphia, Pa. The Westinghouse contract is among the largest awarded for generating equipment in several years.

The single cylinder generating set will consist of a 1800 rpm. turbine, as powerful as any yet built, and a 165,000 kw., 3 phase, 60 cycle, 13.8 kv. double winding generator. The use of the best materials available, special ventilation, and expert coordination of design details will make this unit the most advanced of its kind in the world. About two years will be required to complete the installation; the new equipment is to be ready for commercial service by November 1, 1934.

The Philadelphia Company estimated that eight hundred men eventually will be employed directly on the construction project—the number reaching 100 at the end of this year, 300 in the early part of 1933, 500 by the latter part of 1933; and 800 in the

early part of 1934. In addition to this, there will be the employment given workmen from the manufacture of articles needed for installation and for the production of material to be used.

Of the total eight and one-half million dollars—it is estimated that $2,758,000 will be expended directly on labor by the Philadelphia Electric Company. Much of the remaining $5,472,000 will go into finished material and equipment.

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A new all welded 60,000 sq. ft. warehouse will be erected by the Westinghouse Electric and Manufacturing Company at their Mansfield, Ohio Works for storing electric ranges and refrigerators according to J. S. Tritle, Vice President and General Manager. Work is to start in July and the four-stories-high building will be completed by November 1. The cost is estimated at $150,000.

Built of brick and steel, the large building will be thoroughly modern and will contain three crane aisles, each 65 feet wide, in which stock can be piled 40 feet high.

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The Ferris Wheel of the World's Columbian Exposition of 1893 and the Eiffel Tower of the Paris Exposition of 1900 will have their counter-part in the "Sky Ride" of Chicago's 1933 Exposition.

Contracts have been signed to erect on the grounds of A Century of Progress Exposition, two towers each 600 feet high, 2,000 feet apart and connected by cables carrying cars at the two-hundred foot level, according to an announcement by officials of the Exposition. The attraction is designed to give next year's visitors the supreme in the realm of thrills. The "Sky Ride" is expected to cost approximately $1,000,000. It will be a unique engineering and construction feature of the Exposition.

Two thousand tons of steel are required for the towers. Another thousand tons of steel will go into the sixteen cables—eight for the two aerial tracks and eight for the supporting cables, including the 600-foot backstays.

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WITH THE MANUFACTURERS

The Nela Park Laboratories of the General Electric Co. are responsible for a new development in high-wattage lamp construction which is a radical departure from traditional design.

Ever since Edison made his first lamp, the current has been introduced into the lamp by means of lead wires sealed into glass. In the new simplified construction, however, the leading-in wires are fused to metal, thereby eliminating what was once a center of weakness, and making possible lamps that are inherently stronger than their predecessors.

Essentially, the ordinary incandescent lamp consists of a glass stem structure which carries the leading-in wires and the filament. Around this structure and sealed to the glass stem is a bulb, and to the bulb a base is finally added making connection with the leading-in wires.

The new problem of high-wattage lamp design has now led to a simplified construction using fewer parts, and changing the entire operation of lamp fabrication. Starting with two copper prongs which serve as a base, and to which a special heat resisting glass cup is sealed, the entire internal structure is built up from the prongs, and the bulb is sealed to the glass cup as a final operation.

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A constant-tension regulator has been developed by the Reliance Electric & Engineering Co., Cleveland, Ohio, manufacturers of alternating and direct current motors. This is for use with devices such as reels for winding strip steel, copper or brass; winders for paper and cloth; and other applications where it is necessary to have the material wound or pulled at a constant tension.

For any speed at which the material is being wound or treated, the regulator will keep the pulling tension constant. This apparatus eliminates the need for the slipping clutch which has proved troublesome and unsatisfactory for obtaining constant tension. It is easily adjusted and when once set does not change. Maintenance, with the Reliance Regulator, becomes negligible.

The regulator is very sturdy in construction and liberal in electrical design. Ball bearings are provided. The rheostat used is especially built for this work.

Reliance Constant-tension Regulators may be furnished in various sizes for the many classes of work to which they are adaptable. They are fully patented.

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The Trumbull Electric Mfg. Co. of Plainville, Conn., have announced an addition to their line of Manually Operated Switches; a 2 H.P. Type "D" Enclosed Switch for general use and motor starting service.

This switch, No. 24921, rated 2 H. P. at 230 V. A. C. or 250 V. D. C. is of 30 Amp. capacity, fusible and has 2 Poles and 2 Blades.

The blades are of the double butt wiping contact type operating under pressure.

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OBITUARIES

Gilbert Follansbee, general manager of the Toronto, Ohio, plant of the Follansbee Brothers Co., Pittsburgh, died at his home in Steubenville, Ohio, on July 13, following an operation. He was born at Pittsburgh 45 years ago, and was the son of William U. Follansbee, chairman of the board of the Follansbee company. He had been identified with the company during his entire business career.

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Col. Edward Hall Williams, president of the Vulcan Mold & Iron Co., Latrobe, Pa., and for 40 years prominently identified with the development of the iron and steel industry, was killed in an automobile accident near his home on July 7.

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Thomas J. Foster, 70, assistant master mechanic of the National Tube Co., McKeesport, Pa. works, until his retirement three years ago, died at Pittsburgh, July 14.

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E. T. Wood, Metallurgical Engineer of the A. M. Byers Co., Ambridge, Pa., died on August 2 in Steubenville, Ohio. Until recently Mr. Wood had been Metallurgical Engineer at the Steubenville Plant of the Wheeling Steel Corporation.